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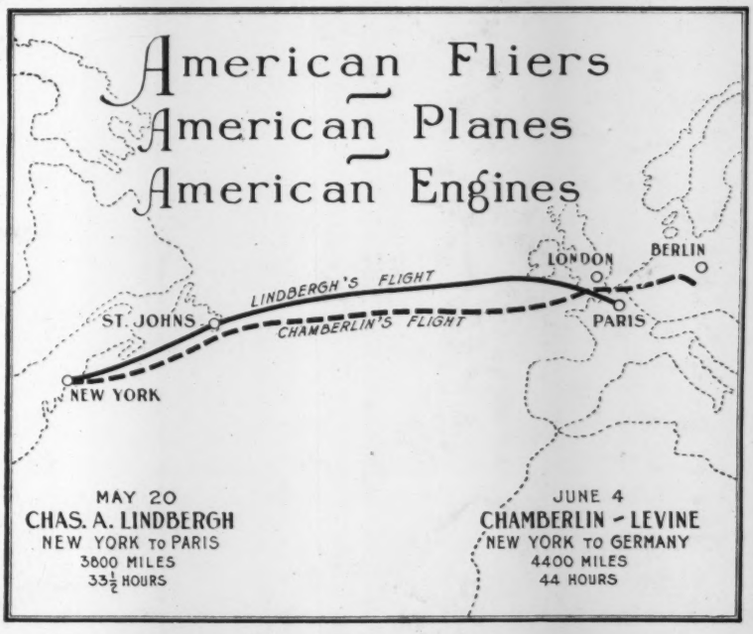

JULY 1927

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Your neighbor the petroleum refiner adopts the oxy-acetylene process*

IN 1913 about 11% of the crude oil was recovered as gasoline. Today the gasoline yield is about 37%. Back of this amazing development are a number of factors and one of the most essential of these is oxwelding.

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* No. 7 of a series of advertisements on the engineering phases of oxy-acetylene welding and cutting. Send for the booklet entitled "Engineering and Management Phases of Oxwelded Construction."

Mechanical Engineering

The Monthly Journal Published by

The American Society of Mechanical Engineers

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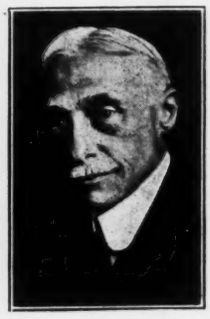
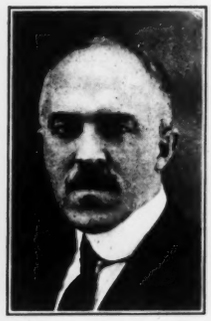
Earle Buckingham

H. F. Guggenheim



E. A. Sperry



R. C. H. Heck

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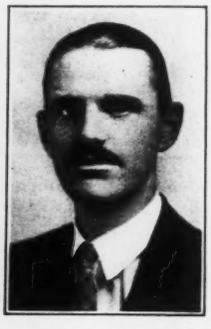

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MECHANICAL ENGINEERING

Volume 49 July, 1927 No. 7

Importance of Aerodynamic Safety in Aviation

What Is Meant by Safety in Aviation—Flying Hazards and How They Are Being Overcome—The Daniel Guggenheim Safe Aircraft Competition and Its Object—Safety Tests and Demonstrations of Competition and Basis of Award of Prize

By HARRY F. GUGGENHEIM,[1] NEW YORK, N. Y.

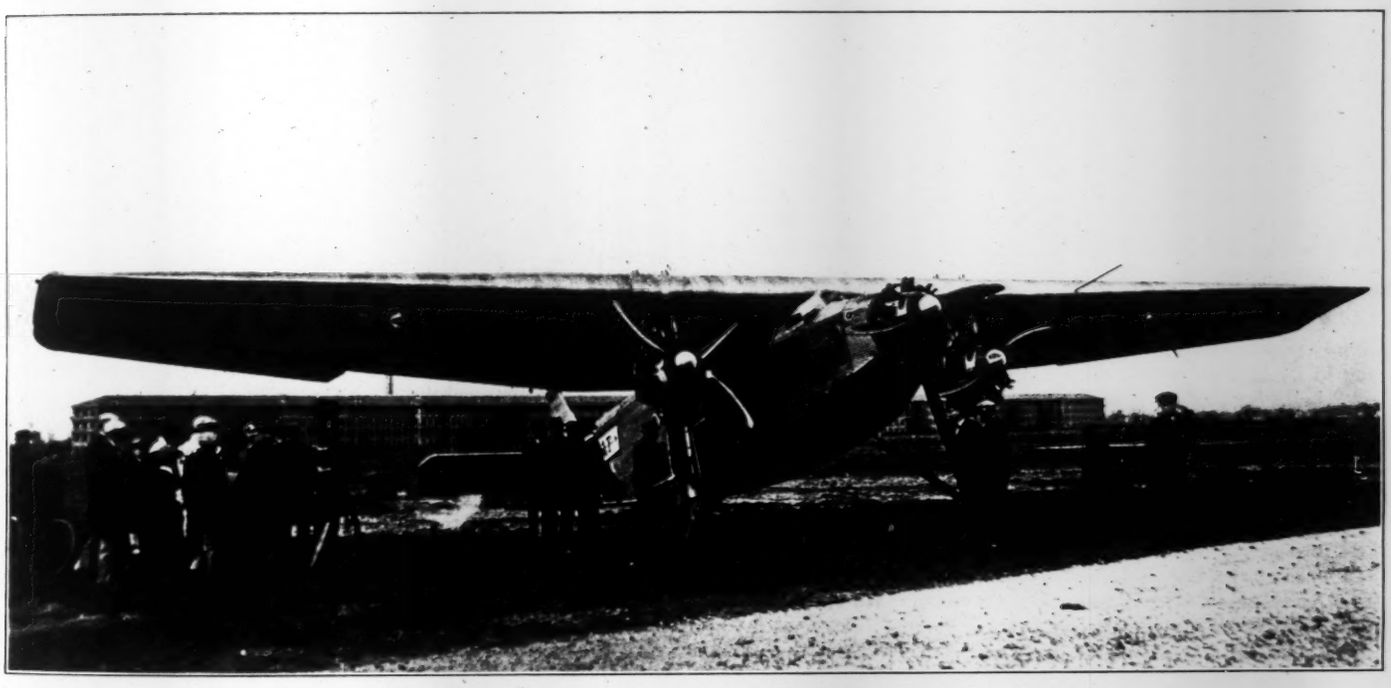

FIG. 1 THE FOKKER 8-PASSENGER PLANE WITH 3 WRIGHT ENGINES OF 200 HP. EACH
(This plane has remarkably good control at the stall without the aid of any special devices.)

"THE largest item in railway returns bids fair to be the list of killed." This quotation from the press of 1845 is borrowed from a recent report of Sir Samuel Hoare. Such a statement from the past considered in the light of subsequent events is a pertinent lesson from human experience that may well be studied by present-day aviation skeptics. No one who flies today under proper organization and circumstances takes unusual risks as we know them in modern civilization. With further development, flying can be made at least as safe and as popular as motoring.

Unfortunately there seems to be a tacit agreement among people interested in aviation not to discuss the dangers of flying. Or if the sacred subject must on rare occasions be examined, the results are the carefully guarded arcana of aeronautics. This traditional taboo of the subject of safety probably can be traced to a desire on the part of aviation's early priesthood to protest against the existence of dangers which they had no means of overcoming. The author can conceive of no more stupid policy than continuing this traditional silence which has left the public so completely ignorant of the cause of aviation that it grossly exaggerates these dangers and ignores the elements of safety in flying.

Recently the author took part in an aviation program broadcast over the radio. He mentioned the subject of safety, and at the end of his address was immediately asked in a tone of amazement by a reporter if he implied that aviation was not safe! Being merely a heretic and not a fanatical heretic, the author assured him that flying was far safer than the public believed it to be. The reporter's only amazement of course was at the author's mention of this forbidden subject. He knew full well that there was hardly a reader of his paper who did not consider flying a synonym for neck breaking. Not long ago the author was amazed in talking to a friend (a usually well-informed man) to find that he believed that upon the failure of the motor in an airplane a crash was inevitable; and probably there are still in this country hundreds of thousands of people who have the same impression. This misapprehension on the part of the public cannot, the author thinks, be overcome by any amount of shouting that aviation is "perfectly safe;" it must be overcome by giving the public the knowledge of aviation which today they so woefully lack. And one way to enlighten the public is through a full and frank discussion of everything about aviation, including its dangers and the manner in which they have been and are being overcome.

The object of this paper is to discuss frankly some of the problems of safety in aviation. There seem to be two opposed schools of thought in regard to flying. One of them asserts that flying is safe and quotes some very conclusive statistics of mileage flown

[1] President, The Guggenheim Fund for Aeronautics.

Presented at the National Meeting of the A.S.M.E. Aeronautic Division, Buffalo, N. Y., April 25 and 26, 1927.

over certain passenger and mail routes with an extraordinarily low fatality.

For example, the British Air Transport Services during the last seven years have had only four accidents involving the death of passengers in flying 5,000,000 miles. Again in 1926 the companies now composing the Luft Hansa, a merger of 54 air lines in Germany, carried 56,268 passengers and 1,888,000 pounds of mail, freight, and baggage for a total of 3,838,425 miles with only one fatal accident. In this country the national airways, for instance, which are maintained by the Army Air Corps and which extend, roughly speaking, from New York west to Kansas City and south to San Antonio, have been traversed during the four years since their establishment by planes traveling a total of 1,200,000 miles, or forty-eight times around the earth. During these four years there has been but one serious accident. An impressive record has also been made by the torpedo- and bombing-plane squadron of the United States Navy. The twelve planes of this squadron, which was organized over a year ago, have flown 216,000 miles, or equivalent to eight and a half times around the world, without a single serious accident.

The other school of thought in regard to flying points to the inherent aerodynamic shortcomings in the modern airplane and contends that until these qualities are improved flying cannot be safe. Their case may be thus stated:

Today, when the speed of the conventional airplane is reduced below its so-called "stalling speed," it ceases to function normally. If the airplane is flying at sufficient altitude and is aerodynamically well constructed, the pilot may recover from the "stall" and regain control. However, should the stall occur in proximity to the ground, the cause of a great number of accidents, a crash is inevitable. In a like manner in forced landings, crashes are commonly due to the high speed necessary to avoid a stall in attemping to fly over some object adjacent to a closely confined landing area or in landing on uneven or restricted ground.

What Is Meant by "Safety in Aviation"

Many differences of opinions can be traced to inadequate definitions, so that it may be well to consider here what is meant by "safety in aviation." For our purpose, which is not theoretical but intensely practical, safety in travel is a condition of minimum risk such as has been achieved by our best railroad and steamship lines. The public draws a comparison between these standard means of transportation and aviation, but like most questions involving human speculation, there are other considerations than safety which play a conscious or unconscious part in any such comparison. For example, consideration is given to the speed advantages of air transport, inasmuch as the history of transportation indicates human willingness to take a *reasonable* risk for increased speed. On the other hand, factors such as reliability, comfort, and price of service play an important part in the public's appraisal of the utility of aviation, which is unconsciously considered in conjunction with its safety.

The contention is often made that any definition of safety must take into account the skill of the pilot. In whose hands is the aircraft safe? Inasmuch as our definition is in terms of comparison with other means of transportation, we must assume that in all cases the vehicles are in expert hands. An airplane is dangerous in the hands of an inexperienced pilot, but so also is a train, ship, or motor car in the hands of a novice.

Those who point to the inherent shortcomings of present-day aircraft state that skill for "safe operation" (the term always used comparatively) is beyond the ability of man. Whether or not this is a fact, there are some who conceive a new generation brought up from early youth in the air who will possess a skill in aircraft operation that has become second nature. To them a landing, which to the author always requires care and attention, will be an unconscious action similar to walking down stairs. There does not appear to be much justification for this conception, heartening and pleasurable though the acceptance of it would be. There is not, the author thinks, any analogy to be derived from human experience to justify the conclusion. On the contrary, for example, man has followed the sea from time immemorial, but to come safely alongside of a dock in so simple a craft as a rowboat has never become an unconscious act and still requires thought and care.

Hazards of Aviation

Let us now consider in what the hazards of aviation consist, and attempt to analyze them. They can all be traced to physical sources. The fundamental problem in flying has been to counteract the law of gravity. But since the Wright Brothers showed this to be feasible, other physical conditions have had to be considered which have added greatly to the complexity of safety problems in practical aviation. The additional physical conditions are the nature of the terrain, clouds, darkness, wind, and sleet.

These primary physical difficulties are being overcome by two means: (1) organization and skill of operation, and (2) improvement of equipment. Champions of each means stress their respective importance. There is nothing, however, fundamentally antagonistic between one and the other, so that accepting one does not mean rejecting the other. Even though one may be firm in the conviction that equipment must be improved, we need still not overlook the necessity for organization and skill in aerial transportation comparable in a measure to that now existent for rail and water transportation.

Flying operations are daily making great progress in the training of personnel, in the establishment of airways and airports, the perfection of ground organization, and lastly in the improvement of communication and meteorological services in which there still remains so much to be accomplished.

There has likewise been great progress in the improvement of equipment, particularly in power-plant reliability, structural safety, removal of fire hazards, and development of auxiliary flying equipment.

Progress Made in Improving Equipment

Although time will not permit a discussion of many interesting and important developments under these headings, in passing one should not fail to mention in particular the developments of multimotored aircraft able to fly with disability of one or more motors, and the amphibian airplane with its wider scope for safe landings under certain circumstances. In improvement of equipment there are several almost virgin fields awaiting cultivation, viz., (1) aircraft with motors accessible in flight; (2) fireproof and shock-absorbing aircraft in which crashing will not entail injury to the occupants; and (3) instruments to prevent collision in the air.

Important progress has also been evidenced in many excellent designs by both American and foreign manufacturers in the aerodynamic qualities of aircraft, but this progress has been primarily directed toward efficiency rather than safety. In some of the basic features of aerodynamic safety, the design of aircraft in use today with some notable exceptions, has not progressed much since the memorable flight of the Wright Brothers at Kitty Hawk.

The reason for lack of progress in this direction seems quite obvious. Nearly the entire impetus to aviation development has been given by governments for war purposes. Consequently a premium has been put on design in aircraft for speed, maneuverability, and other military desiderata, while aerodynamic safety, which is of a secondary consideration in war, has been neglected. Since the war, the infant airplane industry in all parts of the world has not prospered. Manufacturers are unable to develop on their own initiative types that have not a definite sales value. This restricts development for the most part to war types. Some orders, to be sure, are placed for commercial planes to operate over government-subsidized air routes in Europe and for our newly organized mail routes and aerial-service operations in America, but this demand has hitherto been insignificant in comparison with the military demand.

Such commercial planes as are made are naturally adaptations by the manufacturers of military types to meet commercial needs. The infant air-transport and aerial-service companies are struggling against financial adversity. They are unable to insist on the special design of experimental aircraft which shall show exceptional qualities of aerodynamic safety. Their present most urgent need as a matter of self-preservation is a plane with a high "pay load," economical of operation and construction.

In spite of these difficulties in the art, the science of aeronautics has been making important strides. From fundamental research work now in progress and from the construction of some experimental types of airplanes involving a number of radical aerody-

namic departures from the conventional airplane, sufficient progress has already been made in the direction of aerodynamic safety to justify faith that a complete solution of this problem is possible and that it may even be very near at hand. For example, great progress has been made recently toward the solution of the problem of complete control at low speeds by the aid of such devices as Mr. Handley-Page's slot and aileron control. Captain G. T. Hill's tailless airplane is a highly important step in the direction of inherent stability. Senor de la Cierva's Autogiro is an extraordinary development pointing a way toward aerodynamically safe flight in which new and interesting principles are employed.

With the conviction of the necessity and feasibility of aerodynamic safety in aircraft, the Daniel Guggenheim Safe Aircraft Competition has been conceived. The Competition has been framed to focus attention on and elicit from American and other inventive genius aerodynamic improvements in aircraft from the safety point of view.

Essential Characteristics of Present-Day Aircraft

The essential characteristics of present-day aircraft which require improvement have been admirably summarized by Major

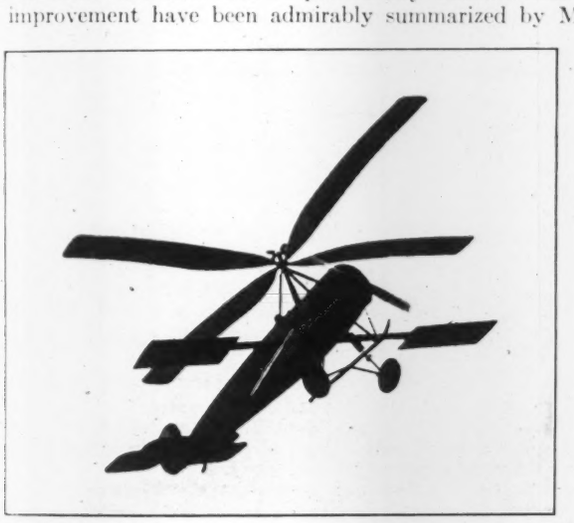

FIG. 2 LA CIERVA'S AUTOGIRO

(A freely rotating giant windmill is used instead of a fixed sustaining surface. Very low landing speed and steep descent are secured.)

R. H. Mayo, representative of the Fund in England, as follows:

1 The landing speed is far too high, and the length of run after landing is too great.
2 The gliding angle is too flat, making the approach to a given spot for landing too difficult.
3 The length of run required before taking off is too great.
4 The angle of ascent after taking off is not great enough.
5 If the airplane is stalled, it becomes unstable and at the same time control is lost.....

The rules of the Competition, drawn to encourage the development of design in aircraft to overcome these disadvantages, have been drafted with the collaboration of aeronautical experts in many parts of the world. A preliminary and tentative draft of the rules was made in the first instance with the technical advice of Prof. Alexander Klemin and Major R. H. Mayo, assisted by Archibald Black. In addition in their final preparation, through the courtesy of Assistant Secretary of War Davison, Lieut. E. E. Aldrin was made available for valuable assistance. Also special consideration from the pilot's viewpoint has been given to the rules by Major R. W. Schroeder, Lieut. James Doolittle, and J. B. Hill, veteran air-mail pilot.

Grateful acknowledgment is due to these gentlemen for their earnest and splendid contributions to the cause of this Competition. Because of their number it is impossible to mention the names of those organizations, manufacturers, engineers, pilots, and scientists in America and Europe through whose invaluable advice and constructive criticism the completion of these rules has been made possible. Perhaps their efforts will bear fruit by inspiring progress through the very suggestiveness of the rules.

Object of the Daniel Guggenheim Safe Aircraft Competition

The object of the Competition is to achieve a real advance in the safety of flying through improvement in the aerodynamic characteristics of heavier-than-air craft, without sacrificing the good practical qualities of the present-day aircraft.

In addition to a statement of the terms and general conditions, the rules consist of two parts, (1) Qualifying Requirements, and (2) Safety Tests and Demonstrations. The qualifying requirements have been drawn to prevent "sacrificing the good practical qualities of the present-day aircraft." The second part, Safety Tests and Demonstrations, is all that the author will present on the present occasion.

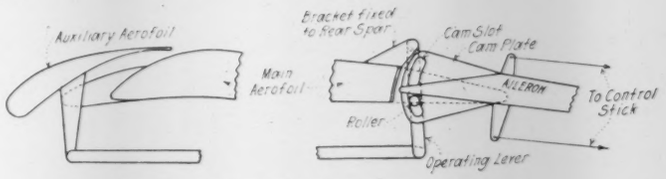

FIG. 3 THE HANDLEY PAGE LATERAL CONTROL

(In this an aileron works in conjunction with a front, movable, auxiliary airfoil or slot. Excellent control at stalling speeds is secured by this device.)

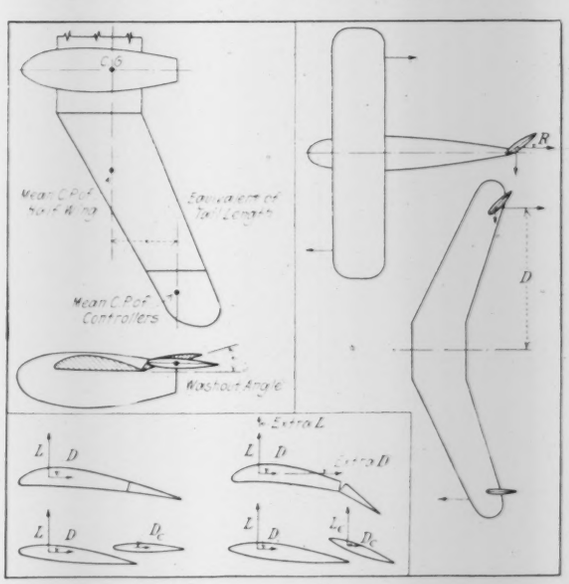

FIG. 4 DIAGRAM OF HILL'S TAILLESS AIRPLANE

(By means of reflexing of the trailing edge and the sweepback, a constant position of center of pressure is secured. Instead of the conventional ailerons, controllers, are used at the wing tips, so balanced statically as to light always in the line of flight. These controllers are therefore perfectly effective when the wing is at its maximum angle of incidence. Hill's machine can descend steadily at an angle of incidence of 45 deg. to the flight path.)

There are altogether nine tests, some containing several parts, as follows:

1 Speed Tests
2 Test of Landing Run
3 Test of Landing in Confined Space
4 Test of Take-Off
5 Test of Gliding Angle
6 Test of Stability in Normal Flight
7 Test of Ability to Recover from Abnormal Conditions
8 Test of Controllability
9 Test of Maneuverability in Restricted Territory and on the Ground.

The object of the tests and the requirements are as follows:

SAFETY TESTS AND DEMONSTRATIONS OF GUGGENHEIM SAFE AIRCRAFT COMPETITIONS

1—Speed Tests

Object: To demonstrate the ability of the aircraft to fly and glide at much lower speeds than is possible in the case of present-day commercial aircraft, thus reducing the risk involved in negotiating forced landings, particularly under conditions of bad visibility and when approaching small and confined landing space.

Requirements:

a Minimum Flying Speed. The aircraft to maintain level and controlled flight at a speed not in excess of 35 m.p.h.

b Minimum Gliding Speed. The aircraft to be able to glide for a period of 3 minutes with all power switched off, during which time the air speed shall never exceed 38 m.p.h.

The aircraft will not be considered to have passed either of the above tests unless it is clearly demonstrated that each and all of the controls are properly effective at the minimum speed specified.

2—Test of Landing Run

Object: To demonstrate the ability of the aircraft to effect a safe landing in a small field.

Requirements:

The aircraft shall land with all power switched off, and after first touching the ground shall come to rest within a distance of 100 ft.

The landing shall be made in a straight line; turning, side slipping, or trick flying will not be permitted.

The use of braking devices will be permitted provided that control is fully retained until the aircraft has come completely to rest, and provided that no serious injury to the surface of the landing field results.

Such braking devices must not require special equipment which is not carried on the aircraft in flight.

3—Test of Landing in Confined Space

Object: To demonstrate that in case of complete engine failure the aircraft can approach and land in a small confined space surrounded by obstructions, such as trees, buildings, etc.

Requirements:

The aircraft shall make a steady glide in over an obstruction 35 ft. high and land in a straight line with all power switched off. After landing, the aircraft shall come to rest within a distance of 300 ft. from the base of obstruction.

The approach to the landing ground shall be straight; turning, side slipping, or trick flying will not be permitted.

The use of braking devices will be permitted under the conditions set out under (2) above.

4—Test of Take-Off

Object: To demonstrate that the aircraft can take off from a small field and after taking off can climb at a steep angle so as to clear obstructions, such as trees, buildings, etc.

Requirements:

a The aircraft shall take off after running not more than 300 ft. from a standing start. The aircraft will not be considered to have passed this test if it touches the ground again after taking off.

b After taking off within a distance of 300 ft. from a standing start, the aircraft shall clear an obstruction 35 ft. high at a distance of 500 ft. from rest. The approach to the obstruction shall be straight and trick flying will not be permitted. External assistance in starting the run will not be permitted in either *a* or *b*.

5—Test of Gliding Angle

Object: To demonstrate the ability of the aircraft to glide for a reasonable distance in case of engine failure and alternatively to glide at steep angle in order to facilitate the approach to a possible landing ground.

Requirements:

a Flattest Glide: The aircraft shall be able to glide with all power switched off so that the angle between the flight path and the horizontal is not greater than 8 deg.

b Steepest Glide: The aircraft shall be able to glide with all power switched off so that the angle between the flight path and the horizontal is not less than 16 deg. During this test the air speed shall not exceed 45 m.p.h. In both cases the aircraft must demonstrate that all the controls are definitely effective throughout the test, and that it can land safely out of this glide from a useful altitude.

6—Test of Stability in Normal Flight

Object: To demonstrate that the aircraft is stable under all normal flying conditions, that is to say, that if the attitude of the aircraft is disturbed either by gusts or by the application of the controls, the aircraft shall return to its original attitude of its own accord when the controls are left free.

Requirements:

a Longitudinal Stability: The aircraft to be provided with means by which it can be trimmed so as to fly with the elevator control free at any speed within the range of 45 m.p.h. to 100 m.p.h. and at any throttle opening of the engine or engines. The test of longitudinal stability shall be as follows:

The elevator control to be moved toward its maximum extent either backward or forward sufficiently to give a fair test of stability and then released. In either case the aircraft must return to steady flight in its original attitude within a reasonable time.

b General Stability: The aircraft to be capable of flying at any air speed from 45 m.p.h. to 100 m.p.h. and at any throttle opening of the engine or engines with all controls left free for a period of not less than 5 minutes in gusty air.

In the case of a multi-engine aircraft, all the engines may be throttled to the same extent.

During any of these tests the aircraft must hold a reasonably steady course, and if disturbed from its normal attitude, must return to such attitude within a reasonable time and without losing height appreciably.

7—Test of Ability to Recover from Abnormal Conditions

Object: To demonstrate the ability of the aircraft to recover from any attitude into which it may get either because of disturbances in the air or of incorrect application of the controls by the pilot, or of sudden failure under difficult circumstances.

Requirements:

a Test of Ability to Maintain Control in Case of Engine Failure

The aircraft must demonstrate that it can be satisfactorily controlled by the pilot in case of sudden engine failure when flying at any attitude. In case of multi-engine aircraft, the pilot must be able to maintain control when any one of the engines or any combination of the engines or all of the engines suddenly fail.

The aircraft will be required to demonstrate that in case of complete engine failure (all power being switched off) it will take up a steady gliding attitude when all controls are left free from the moment of switching off the engine or engines.

The aircraft will further be required to demonstrate that if the elevator control is pulled in toward its maximum extent at the moment of switching off and held in that position, the aircraft will remain under control, not get into any violent or dangerous maneuvers, and descend on a steep glide path at a speed not exceeding 40 m.p.h.

b Test of Ability to Recover from Violent Disturbances

1 The aircraft to be dived with all power switched off until air speed reaches 20 per cent above maximum level flying speed. At this speed it must answer all the controls correctly and effectively. All controls will then be released and the aircraft must of its own accord return to a steady gliding attitude without serious loss of height.

The aircraft may be required to pass this test when trimmed for any speed from 80 m.p.h. to 110 m.p.h.

2 The aircraft to be flown at full throttle at an air speed of 45 m.p.h. trimmed for any speed from 45 m.p.h. to 75 m.p.h. The engine is to be switched off and at the moment of switching off, the pilot is to move any one or any two, or all of the controls in such manner that an abnormal attitude results. Complete recovery is to be made with the aid of the controls and a steady glide to be taken up with a loss of height of not more than 250 ft. from the height at which the abnormal flight attitude was obtained.

The same test is repeated with the exception that when in an abnormal flight attitude, all controls are to be released, and the aircraft must make complete recovery of its own accord and take up a steady gliding attitude with a loss of height of not more than 500 ft.

8—Test of Controllability

Object: To demonstrate that the control system is simple and easy to operate and that under all conditions of flight and gliding, effective control in all senses is maintained. Every aircraft must be equipped with three substantially independent controls corresponding to three axes mutually at right angles, and these three controls must continue to be substantially independent and collectively effective in the same direction at any attitude of the aircraft from the attitude at maximum level flying speed to the attitude at the steepest glide and at any throttle opening of the engine or engines. The aircraft must clearly demonstrate that it is not subject to complete loss of control when any particular attitude is reached, such as the loss of control which accompanies the phenomenon of stalling.

Requirements:

The aircraft will be required to demonstrate the effectiveness of each and all of the controls at any speed and at any attitude from the attitude at maximum level flying speed to the attitude at the steepest glide and at any throttle opening of the engine or engines. This will be tested by making definite and sharp movements of any particular control or controls, which must result in the aircraft making corresponding definite rotations about the respective axes. The controls must not only be effective in producing disturbances of attitude, but also in enabling rapid and definite recovery to be made from disturbances.

The effectiveness of any one control or combination of two controls when the other control or controls are left free or held in fixed position must be demonstrated.

The aircraft may be called upon to pass these tests in calm or gusty air.

9—Tests of Maneuverability in Restricted Territory and on the Ground

Object: To demonstrate that the aircraft can be safely and effectively maneuvered when taking off or landing in restricted territory and when taxiing on the ground under its own power without external assistance.

Requirements:

a Maneuverability in Restricted Territory

A squarep lot, 500 ft., by 500 ft., will be marked off and shall be considered as surrounded by an obstruction 25 ft. high along its entire boundary. The pilot shall take off in any manner he judges best, and climb either above the square plot or outside of it, providing he passes above the imaginary boundary obstruction. The engine may be switched off at any time and the pilot shall land the aircraft within the square plot without passing through the imaginary boundary obstruction.

b Maneuverability on the Ground

The aircraft will be required to demonstrate that it can be taxied under its own power and without external assistance in any direction in a wind whose mean speed at ground level is at least 20 m.p.h.

The aircraft will also be required to demonstrate that it can be easily

handled and moved by ground personnel as required of any aircraft operating for commercial purposes.

Basis for Award of Prize

The winner of the Competition will be the competitor whose aircraft satisfies all the Qualifying Safety Requirements and in addition is awarded the highest number of points in accordance with the schedule in tests 1 to 4. (See next column.)

These rules undoubtedly have their defects: perfection is hardly to be expected in so difficult and intricate a problem. If it were an easy one, there would be no need for the Competition. The rules should be considered as a whole and due allowance made for the interdependent integral parts, some of which were necessarily formed by compromise.

To return to our definition of "safety," if aviation is dependent upon the public's opinion of the safety of flying and the author for one is convinced that it is, perhaps this competition will be helpful in showing how very much safer flying is than the public realizes, and as well, he hopes, in proving not by statistics but by actual demonstration that airplanes can be made inherently as safe as steamships or railway trains.

When the public is firmly convinced of this, aviation will assume almost over night its proper place in the front rank of transportation. Let aviation gain the confidence of the public and the economic problems will solve themselves.

The aviation industry is unfortunately today traveling in a vicious circle. It is not prosperous through lack of public confidence in flying, and public confidence is difficult to win without financial resources that come with prosperity. The Daniel Guggenheim Safe Aircraft Competition, the author sincerely hopes, will act as a tangential force helping to break this vicious circle into a straight line of progress.

Qualifying Safety Requirements

	Maximum number of points obtainable
1—*Speed Tests*	
a 2 points for every m.p.h. less than 35 m.p.h. at which level controlled flight can be maintained	10
b 4 points for every m.p.h. less than 38 m.p.h. which is not exceeded in a steady controlled glide during a period of 3 minutes	24
c Any aircraft which obtains a combined total of at least 24 points under tests *a* and *b* will be eligible to receive points for high speed in excess of 110 m.p.h. as follows: 1 point for every 2 m.p.h. in excess of 110 m.p.h. at which level flight can be maintained	10
2—*Test of Landing Run*	
2 points for every 3 ft. less than 100 ft. in coming to rest after first touching ground	40
3—*Test of Landing in Confined Space*	
1 point for every 2 ft. less than 300 ft. from the base of an obstruction 35 ft. high in coming to rest after gliding in over the obstruction	75
4—*Test of Take-Off*	
1 point for every 15 ft. less than 300 ft. required to take off from standing start	15
1 point for every 10 ft. less than 500 ft. to clear obstruction 35 ft. high from a standing start	26
	200

The Light Supercharged Diesel Engine for Use in Air Service

Particulars of Author's Proposal to Reduce Diesel-Engine Weight per Horsepower Below That of Gasoline Engines by Employment of High Supercharging, and to Eliminate Fire Risk from Flying by Use of Diesel Engines Running on Cheap, Low-Volatile Fuel Oil

By ELMER A. SPERRY,[1] BROOKLYN, N. Y.

THE Diesel engine has always been notoriously heavy. The maker who has equipped more ships with them than any other started out with an engine weighing about 450 lb. per hp. A new design which weighs somewhat less than this is still extremely heavy. Starting at this point, the best minds in automotive engineering have been endeavoring to reduce this weight so as to approach the point when we can commence to think of adopting it for aeronautical service.

For airplane service, for instance, the engine is only one of the elements of weight that has to be considered in connection with the whole power-plant equipment. The fuel must also be given serious consideration, because the plane has to get off with fuel enough to take it to its journey's end. In the words of a technical department head of the British Air Ministry, "The effective weight of the power plant for an airplane which has to get off the ground fully equipped with fuel is the total weight of the engine plus that of the fuel. Any reduction in the fuel weight necessary for a given journey may be set off against any increased weight of the engine itself. The longer the journey, the more will high fuel economy compensate for a heavier engine.... The compression-ignition engine of 3 lb. can compare with a petrol engine of 2 lb. per hp." The great factor that comes in just at that point is that the Diesel cycle with its higher pressures and expansion ratio is able to deliver the same horsepower, in the words of the above authority, "at considerably lower fuel consumption." So if the engine itself can be placed on an equal footing as to weight per horsepower, an advance will be made that outclasses anything possible with the petrol or gasoline engine, and which in addition will (1) eliminate all fire risk by use of non-inflammable fuel oil, contributing immeasurably to safety, and (2) employ a fuel costing about one-tenth what aviation gasoline does, volume for volume.

What Has Been Done and What Is Still Possible in the Way of Reducing the Weight of the Diesel Engine

If this can be accomplished it is of such extraordinary importance that we can afford to go into detail as to what has been done and what it is still possible to do in reducing the weight of the Diesel engine so as to bring it within the possibilities of airplane service.

Refinement of Detail. The problem has been attacked by going to extreme refinement of detail, reducing each part and element to the ultimate in size and weight while still retaining sufficient strength. On this last point of strength it is proper just here to correct a popular misunderstanding. The gasoline engine in which the pressure rises at the time of each ignition to a perfectly well-understood maximum, is subject to occasional extreme pressure due to preignition or to detonation of the fuel within the cylinders, and strength must be provided to withstand these extreme pressures. This has now become so well understood that complete provision is made for it in the Liberty, Curtiss, Wright, and practically all of the present-day aviation motors on both sides of the Atlantic. Just here a fortunate circumstance obtains regarding the Diesel cycle. It is entirely free from any detonation possibility, so that when the engine is designed to take the maximum pressures there are no super-pressures that need to be provided for such as this freak peak in the case of the gasoline engine.

Careful calculations and designs show that with the same weight the Diesel cycle will give power about in proportion to the ratio of the net mean effective pressures prevailing in the cylinders of each. In the gasoline engine we have 130 lb. per sq. in. net mean effective

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pressure, whereas in the Diesel, owing to the fact that we have only about 1/25 to 1/50 of the time available in which to get our fuel and the oxygen of our air intimately mixed before ignition, it is found that our combustion is considerably less complete than in the gasoline engine. This of course militates against the Diesel cycle on two counts: (1) we have to increase our lung capacity and therefore our cylinders to accommodate the extra air that never can be reached by the fuel, and (2) our mean effectives are held to the lower value of 75 to 80 lb. So the designer starts out with a twofold handicap.

Facing these facts of more cubic feet of air being required per horsepower delivered on the one hand, and having available much less net mean effective pressure on the other, it is self-evident that the straight four- or two-cycle is not enough. We must look for methods of obtaining more work from each element of the engine if we are to bring the weight within the required limits. Let us now examine in more detail as to what the attempts are that have thus far been made.

The refinements in detail mentioned above have gone forward and been pushed to an extreme under the wonderful stimulus and inspiration of the Air Ministry of England, but the engines developed are heavier than is permissible for airplane service.

Substituting Two-Cycle for Four-Cycle. Another way in which the problem may be attacked is by using two-cycle instead of four-cycle. This is a favorite method in submarine Diesels and in larger marine power plants. It has been found that in these units a saving of some 40 per cent over the four-cycle operation can be achieved. In this two-cycle operation, however, two decided tendencies are developed, namely, (1) to lower the net m.e.p. owing to the impossibility of scouring out all the products of combustion before compression proper must set in for the next cycle, and (2) to smoke when running at even low powers. There is the same drawback of extreme complexity when an attempt is made to make a small two-cycle engine with the necessity of scavenging and charging pistons and other auxiliaries, entailing, in some instances, as high as 10 camshafts and 16 gears for a single unit, and as a final result such engines have shown only a small improvement in weight over the four-cycle unit mentioned above in which extreme simplicity is coupled with great refinement in detail.

Making Engines Double-Acting. Marine engines have been still further but not very markedly reduced in weight by making them double-acting, where both sides of the piston are active but where a piston rod has to pass through heavy packing glands and where intense cooling is required and there is a marked increase of exposure of chilled walls to the raw fuel as well as to the combustion products. The piston rod has to be very large, rendering it practically impossible of consideration for small engines, so that no attempt has ever been made, so far as the author is aware, to adopt this method of reducing weights in either four-cycle or two-cycle operation.

Adoption of Supercharging. The author has reserved to the last the simplest, most promising, as well as the most important of all the attempts, namely, supercharging. It is the most important because here we have an opportunity of overcoming our handicaps by very materially increasing our mean effective pressures and bringing them not only up to an equality with those of gasoline engines but as a matter of fact greatly exceeding them, thus giving a promise of possibly actually excelling the gasoline engine in power per pound. But in this matter of supercharging there are limitations.

The first manufacturers of engines to use supercharging seriously were the Fiat Company in Turin, Italy. So the author's investigations included a recent visit to this company where he was accorded every courtesy possible and discussed this matter with the chief engineer and two of his assistants. Supercharging with the Fiat Company has been for the purpose of increasing the power of racing-type automobile engines, and they have carried it to the very limit of possibility with the ordinary means of control of the elements involved. Their experience is that the principal limitation is the melting of the exhaust valves. They generously gave the author vital details as to where this limitation occurs on the curve of supercharging pressures involved, and as to the technical side of these limitations. At best, the failure is found to occur at comparatively low supercharging pressures. Especially do these pressures seem low when measured in atmospheres.

Strange as it may appear, the failure does not originate from the heat alone. This we know because of the fact that by using some modern highly organized fuels, which might be considered as modified fuels, these precompression or supercharging pressures may be still advanced somewhat, but not very materially. Early in his investigation the author found the temperature alone can almost be considered as a secondary element. The primary element and starting point is the scouring of the valve seats and the impossibility of holding the seal thereafter. Stated in rough terms, the condition is that after erosion has started, both sides of the valve commence to be immersed in the intensely hot gases.

We are required to exhaust into the atmosphere from an exhaust pressure very much higher than normal. These gases stand ready to escape through even the narrowest opening into the atmosphere at incredible speeds. Just at the moment the seal is broken each time the valve starts to rise, which may be many hundreds of times a minute, these excessively high-velocity gases impinging on both surfaces—valve and seat—with their enormous temperatures, carry away small parts of the seat at first until the washing and erosion become very rapid and the seat, and consequent seal, is completely lost. This leaky valve then allows the combustion gases to escape past it at their highest explosion temperatures and pressures rather than at the lower exhaust pressures, the valve meanwhile being completely immersed in the hot gases, which are thus below as well as above it. Therefore the deterioration and destruction of the valve are inevitable and proceed very rapidly.

Serious as the case seems to be, there is really a comparatively simple way to completely overcome this trouble and to push supercharging away past this limitation to still higher pressures than ever heretofore used. The higher pressures render it so completely serviceable as to place the compression-ignition engine, with its high fuel economy due to its pressure and the wealth of its long expansion curve, actually in advance of the low-pressure electrically ignited engine. The actual weight of the heavy-oil engine may easily be less than that of the present gasoline engine, this aside from its complete safety against fire risk, the large saving in weight of the fuel required, and the cost of this lesser amount reduced to measurably one-tenth in dollars, gallon for gallon or barrel for barrel.

Having thus reached a point where we have a clear insight into the various factors of the problem, the author believes we are ready to appreciate that on account of the extreme importance of the objective and the magnitude of the difficulty itself, we are certainly justified in taking any practicable measures necessary for its achievement, no matter how extreme they may at first seem.

There is no problem connected with the piston, cylinder walls, intake valves, or lubrication. All these elements work the same in the supercharging engine as in the ordinary engine. The problem consists in how to overcome the difficulty that the Fiat engineers recently ran into and that the author has realized for many years, as stated, namely, that of holding the exhaust valves. There is no engine that does not cool the exhaust-valve seat, and many well-established methods have been resorted to for cooling the valve itself, so we may dismiss these two items for the moment. But how are we to obtain the seal of the exhaust valve? If this is lost the case is hopeless.

Difficulties Encountered with Exhaust Valve and How They Have Been Overcome

It happened that the author was one of the first to see the great possibilities of supercharging, and the first, so far as available records indicate, to be willing to undertake an investigation of these possibilities under conditions where supercharging was taken seriously and carried to high pressures, for the purpose of discovering the limitations and to start on a campaign looking to their solution. So he built his first highly supercharged engine in 1890 and had it in operation before the World's Columbian Exposition in '93. There was of course no precedent for this work, but the overcoming of the difficulties encountered progressed down through a series of engines that have been built since that early date until some five years ago when, by correcting another remaining fault, quite unexpectedly the valve problem was found to be fully conquered, and the valve has given practically perfect operating results ever since.

The solution consists of not only entire suppression of the gas

velocities at the instant of breaking the seal, but going still farther and preventing any movement whatever of them at this moment. If one will visualize this condition for a moment he will see that the only method of accomplishing this is to do away with the enormous difference of pressure on the two sides of the valve—and never open it except under conditions of equal pressures on both sides. To do this we must provide a vestibule, as it were, to receive the escaping products on their way to the atmosphere. Under these conditions not only no velocities but no movement at all of the gases will occur, the flow being then governed solely by the slow beginning of the movement of the pistons themselves. This would be nothing short of a definite detriment were it not for the fact that means have been found to absolutely reverse this condition and make it furnish us with other definite results of value. Not the least of these lies in supplying all the extra power necessary for the high pressure or supercharging, and, on top of this, actually supplying a large overplus as a definite contribution to the increase of the sum total of motive power of the engine itself.

This vestibule or enclosure has a light piston for the purpose of operating as a cushioning cylinder opposite to the movement of the combustion piston, so arranged as to cup the contained products at a proper point in its upward stroke to bring the cushioning pressures up to equal the pressure of the gases on the other side of the valve, which are the quite high pressures still existing at the lower end of the expansion curve of the highly supercharged combustion cylinder. Cushioning simply acts as a spring, giving back all the power in expanding that it takes to produce the compression.

When supercharging pressures rise to the point where they begin to make a substantial return, the power required to compress the large extra air volume and run it up to, say, 50 lb. gage becomes a substantial amount and must be supplied by the engine. So the cushioning piston is seized upon to supply this extra power. When this is given an area about twice the area of the pump, the gross power thus derived from the second expansion of the products yields about three times the power required by the pump, and we have a dividend of about 70 per cent to hand to the crankshaft which is added to the already extremely high m.e.p. that come from the supercharged combustion cylinder. Taken together when referred back to this cylinder, they are found to make a net m.e.p. of measurably 300 lb. per sq. in., which is enough to easily overcome the three outstanding handicaps under which compression-ignition engines have always been compelled to work. These are: (1) the 80 lb. net m.e.p. as compared with the 130 lb. in the gasoline engine; (2) the greater volume of air required to be handled per net hp. of output owing to less complete combustion, and (3) the somewhat lower speeds demanded by the injection type of engine. It is super-supercharging that has given us this extremely important result. Ordinary supercharging is nowhere near enough.

The next step is simple. By taking two four-cycle engines and operating the cycles in alternate step, one pump cylinder and one cushioning piston are found to do for the two engines on alternate strokes for each. Again, the valve that cups the products at the proper point in the upstroke to produce the cushioning is common for the two engines. The m.e.p. of the pump is about 26 lb. and that of the exhaust gases on twice the area is about 44 lb., giving a 70 per cent overplus of power as the dividend on the credit side mentioned above, and this is delivered as two-cycle twice as often as is power by either of the combustion cylinders. The great gain, however, is in the unprecedented volume of the compression at full Diesel pressures and temperatures, thus making three important gains: (1) cutting down the surface of the chilled combustion-cylinder wall per unit volume and "bunching" this volume, thus giving a decided gain for solid fuel injection over the small—actually minute—clearances of the standard Diesel; (2) gross mean forward pressures never before obtained in compression-ignition engines—on the order of 500 lb. per sq. in. on the expansion curve as compared with about 200 lb. in the gasoline engine; (3) supercharging which gives such a large volume of gases to deal with that the cushioning piston is called into play to recover their value and expand them down nearly to atmosphere, thus adding to the gross mean effective pressures no less than three times the 80 lb. net of the ordinary Diesel combustion cylinder when referred back to this same element. This is purely a by-product recovery from what in the ordinary supercharging engines is discharged at high pressure and velocity into the atmosphere, eroding and ruining the valve seats, as previously described.

The author has stated above that the melting of the exhaust valve by the excessively hot gases in the combustion cylinder has been found to be one of the principal limitations in supercharging engines. This is easily understood by any one who has had opportunity of observing these valves when operating continuously under full-load conditions in a number of different engines of both the Otto and Diesel cycles. These all run at the low "cherry red" and the well-known Liberty engines run regularly at a high-red heat. When studying the Liberty valves under full-load conditions, one wonders why they do not deform—and it is known that they do unless they are built stocky and of a thick "tulip" shape at their centers. The thinner type of exhaust valves, even though of the best alloy steels, soon deform so as to lose their seal by warping, after which the seats erode very quickly and the engine is useless. We thus recognize that it is not a very great step between this condition and the still higher temperatures of the supercharged engines where it is found impossible to so design the valves that they will stand the increased temperatures present. This refers to the higher pressures at the lower end of the expansion curve, due to even a moderate amount of supercharging.

The author has found it very simple to hold these valves down to much lower temperatures than any of those mentioned above and to complete "black" running by the simple act of air washing, subjecting them to the abundant currents of inrushing air from the supercharging-pressure source, be it a piston or rotary compressor. The fact that this air is under superatmospheric pressure multiplies its cooling effect. For instance, the author's 50 lb. supercharging air is about four times as good a cooler, having four times the number of molecules in contact, and the contact intimate. This is to be compared with the sub-atmospheric inhalation air in ordinary motors; all that it is necessary to do is to provide suitable radiating fins on the exhaust valves and guide all the incoming air over these fins.

Advantages Obtained By High Supercharging

In high supercharging the ordinary negative or suction stroke is converted into a positive power stroke with about 50 lb. net m.e.p. delivered to the crank, thus recovering fully two-thirds of the power required by the pump with its mean effective of about 26 lb., but of somewhat larger area. The induction valve is also provided with fins, so it is found a very simple matter to keep the temperature of both of these valves much below the full-load temperatures that generally prevail in exhaust valves of practically every type of engine, be it an aviation, automobile, or a marine engine of either the Otto or Diesel cycle.

In the Otto cycle the time factor for the intimate association of the fuel and oxygen is very long, involving as it does the carburetor and manifold, the inhalation, and the compression stroke—all previous to ignition. Of course no such opportunity exists with the Diesel, so every aid possible must be extended to the fuel in reaching the oxygen after the fuel is introduced. We find here, in comparison with the prevailing Diesel practice, that this extension of time amounts to no less then 300 per cent. The fuel injection is ordinarily cut off at about 32 deg. past the top dead center and the exhaust opens at about 59 deg. before the lower dead center is reached, leaving 89 deg. of crankshaft revolution to represent the time factor for completing combustion in the ordinary Diesel cycle. This has always been a most serious handicap in competition with the gasoline engine, especially when high speed was attempted. Subtracting the sum of the above angles from 360 deg. instead of 180, as we are permitted to do in the supercharging engine, we have 269 deg. which is more than three times the 89 deg. of the standard Diesel. So here is one of the most important incidental contributions on the positive side of the equation, flowing from the adoption of the supercharging cycle with the multiple-expansion feature.

Now let us examine into the slightly increased temperatures that have been received by the incoming air pouring through this valve in the cooling operation, especially as to their true significance in connection with compression-ignition engines. It should be remembered here that we need temperatures as well as pressures for

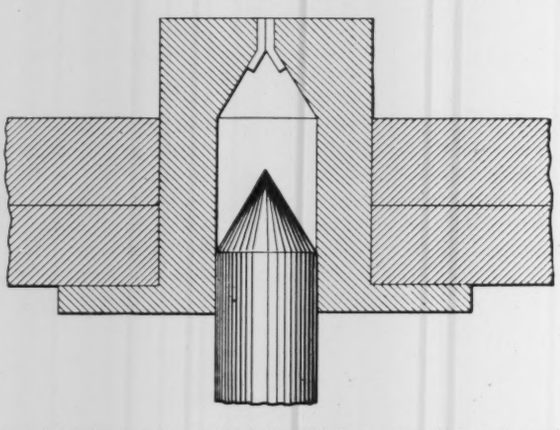
proper ignition at the moment of fuel injection, so even this small amount of heat is not lost. The high pressures are of prime importance in securing the long expansion line which gives us the well-known fuel efficiency of the Diesel cycle over and above that obtained with the much lower pressures of the Otto cycle. While on this point we should note two other advantages. Ordinary supercharging is very wasteful of fuel. The exhaust valve is opened at a high point in the expansion curve, which is thus abruptly terminated, and the large amount of energy represented below this point is thrown out into the atmosphere and wasted. Just here a very great difference exists in choosing the area of our cushioning piston. This piston primarily prevents all high velocities across the valve seat at the moment of breaking the seal and thus saves this serious situation completely. We have redesigned these pistons so they now have an area which is best suited also to complete the long expansion curve, making full recovery of practically all its energy. We are thus able to carry the expansion on down to a point far below that at which the ordinary Diesel or Otto curve stops. This is of course in the interest of still higher fuel efficiency. It is thus found that while still holding the weights to the level discussed with reference to the Otto-cycle gasoline engine, these products can be expanded measurably to 10 or 12 lb. gage before being discharged to the atmosphere, as compared with 40 or 50 lb. in the usual practice, and much higher than this in the supercharged engines. The extremely high mean effective pressures available in this engine easily throw the scale the other way. The difference between the Diesel 75 or 80 lb. net m.e.p. and the 300 lb. produced by high supercharging is sufficient to overcome the handicaps of the 130 lb. of the gasoline engine, together with all the others, leaving us with a definite margin in favor of heavy oil, weight for weight and power for power. Of course there is no complex electrical equipment—magnetos, coils, contactors, distributors, etc.—in the simple compression combustion engine. The ignition is automatic. The complications are thus done away with, together with all the very serious interferences with and disturbances to the most delicate wireless communication. This is another definite advantage.

We have always felt that the compression-ignition engine, notwithstanding its handicaps, when properly organized should outclass all its rivals and come into its own. We feel that the foregoing indicates that we have at last developed that organization which should insure its complete success. With this engine in regular service, accomplishing what has been outlined herein, we believe that it will be found to constitute the one necessary final step that has been awaited to give aviation the greatest single impetus since its beginning. That beginning was in America, and America must also take this advanced step in the interests of safety and economy.

A New Rivet for Aircraft Construction

DURALUMIN is now largely used for aircraft construction, where its high strength and low specific gravity make it particularly valuable. It does not allow of welding, however, and aircraft construction in dural suffers from the disadvantage of an immense number of rivets which sometimes have to be inserted in positions where it is difficult to hammer at one end and support at the other. A new type of rivet invented by Lieutenant J. W. Iseman gives promise of overcoming such difficulties.

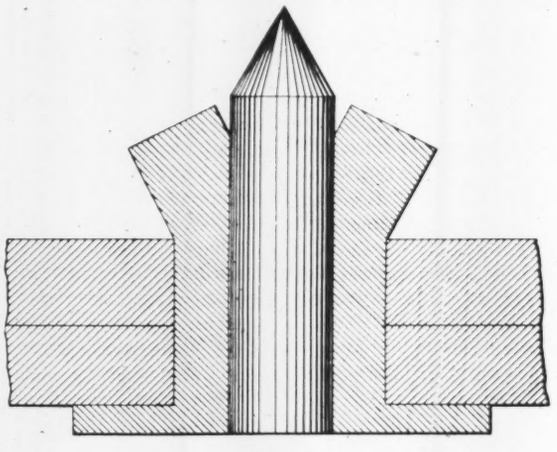
This new rivet is unique in its construction. A hole ending in a conical point is drilled in the direction of the axis of the shank through the head of the rivet. The base of the cone starts at the end of the nominal "grip" (equal to the thicknesses of the riveted plates) of the rivet. The end to be "upset" is slotted (two slots at right angles to each other) up to the nominal grip. A pin—its end also ending in a cone—fits into this hole. Its diameter is the same as the hole; if it were greater it might split the shank of the rivet; if less it would slide in and out too easily.



The rivet holes are drilled, and the rivets inserted. The pin is topped with a hammer until the pin is flush with the head. The pin should be tapped lightly to avoid fracturing the metal when "upsetting." As the pin is tapped, it forces the slotted end to spread out as the straight side of the pin travels along the side of the cone. When the pin is all the way in, the sides of the cone are parallel to axis of the rivet, and are now at an angle equal to the complement of the base angle of the cone. The "forced out" slotted end of the rivet forms an effective "Head."

When the rivet is in place and upset, friction and general binding between the surface of the pin and the rivet proper under pressure is quite sufficient to hold the pin in place, as observation showed.

In an experimental investigation at the Daniel Guggenheim School of Aeronautics, New York University, two strips of brass one-half the length of the rivet "grip" in thickness were riveted together as a lap joint and were pulled apart until the rivet failed in shear. The two free ends were clamped in the jaws of an Olsen testing machine. The rivet failed at approximately 450 lb.



Calculations

Diameter of rivet	0.127 in.
Diameter of pin	0.053 in.
Failed in shear at	450 lb.

$$\frac{\pi d^2 S}{4} = 450 \text{ lb.}$$

$$S = \frac{450 \times 4}{\pi (0.127)^2} = 35{,}500 \text{ lb. per sq. in. shear}$$

The usual value of shear is between 30,000 and 35,000.

Apparent Present Tendencies in Airplane Design

A Brief Review of Recent Developments and Practice in Various Countries, Pointing Out Successful Results Obtained with Designs of Widely Different Types

By V. E. CLARK,[1] BUFFALO, N. Y.

A DISCUSSION of the tendency in airplane design might deal with one or both of two phases: (*a*) the past and present, and (*b*) the future. In this paper the author, lacking the courage to venture predictions, will confine himself to a brief review of the immediate past.

It seems that a fair answer to the question, "In what direction is airplane design leading?" would be: "Scattering in all directions." At no time during the brief history of aviation have there been *successful* airplanes of so many widely differing types—aerodynamic and structural.

One reason for this encouragingly healthy state of affairs may be that more and more engineers are being developed who combine native imagination with a sound knowledge of the pertinent fundamental sciences and a sufficient amount of practical experience (and horse sense) to justify risking the hazards of pioneering, instead of cautiously copying a traditional scheme that has "worked" but which may have become conventional merely because it happened to have been the luckiest result of several early cut-and-try endeavors. One says "encouragingly healthy" with the firm conviction that it would be most unfortunate for us to attempt to standardize at this time, on the assumption that the possibilities of the science have been sufficiently thoroughly investigated and developed to warrant standardization. To be trite, the longer one observes aviation the more one must realize what a tremendous amount is to be learned.

The diversity of development between nations is much greater than that between constructors of any one nation. While there is no defined "school" of design adopted within any one nation, there is consistently more similarity intranational than international.

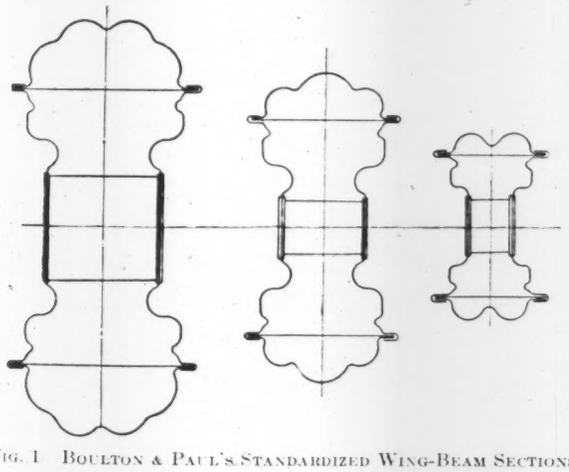


FIG. 1 BOULTON & PAUL'S STANDARDIZED WING-BEAM SECTIONS

One reason for this is the influence of governmental policies as affected by peculiar contemplated military problems and by the availability, among national resources, of special basic materials. Great Britain, for example, must build airplanes which will be durable in all sorts of climates, hot and cold, dry and wet, and which may be maintained in service thousands of miles across the water from their home factories. Other nations lack certain indigenous basic materials. The United States, in this particular, is very fortunate. We have iron, chromium, molybdenum, nickel, silicon, manganese, vanadium, aluminum, copper, magnesium, cadmium, spruce, birch, long-fiber cotton, nitrates, and fuel and lubricating oils. In fact, nature has endowed our country with every element necessary for the fabrication and operation of aircraft. For the few elements we lack, we have acceptable substitutes.

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Presented at the National Meeting of the A.S.M.E. Aeronautic Division, Buffalo, N. Y., April 25–26, 1927.

GREAT BRITAIN

Of the great powers, England is probably the most conservative in aerodynamic design and in the use of materials and manufacturing processes. By far the great majority of modern successful British airplanes are of the type which has flown well for the longest time: the tractor biplane with wings of thin section externally braced, usually with a two-bay cellule and usually of equal spread and chord. Most of the construction is of wood, with fittings of sheet steel. Captain De Havilland, whose DH-4 we know so well in this country, has designed a great number of useful machines, in the general scheme of all adhering to the type of aerodynamic design and of construction which he adopted many years ago. His fuselages are of wooden sticks taking compression and forming the framework for an outside covering of plywood which completes the structure by tying the sticks together and taking diagonal tension loads.

FIG. 2 LA CIERVA'S AUTOGIRO

FIG. 3 VIEW OF HANDLEY PAGE "HAMLET" FOUR-PASSENGER CABIN PLANE, SHOWING OPEN LEADING- AND TRAILING-EDGE SLOTS

In England, welding in vital structures is discouraged. Very little duralumin (we shall use this term for aluminum-copper alloys for convenience at the cost of accuracy) is used, notwithstanding the fact that one of the foremost airplane manufacturers, the Vickers Company, fabricate this material commercially in various forms. The Boulton & Paul Company builds its structures of very thin high-grade steel sheet, with sections so formed and riveted together as to resist local buckling failures.

The development of La Cierva's "Autogiro," invented in Spain, is evidently being encouraged to some extent. In this machine vertical lift is obtained, not by fixed wings, but by four monoplanes

above the fuselage rotating propeller-wise about a vertical axis. Rotation is started by external power before take-off and continues due to the nature of precession of air forces, during revolution, on this peculiar airscrew, as long as there is relative air velocity. The machine is propelled through the air by a conventional tractor propeller-motor unit mounted in a standard fuselage with tail surfaces. Due to relatively high "tip" velocities of wing blades, the airplane can take off and land at comparatively slow true air speed, with resulting short run. The gyroscopic effort of the mass of the rotating wings, with large diameter, tends to stabilize the machine in the air. It is very doubtful, however, whether this machine will compare favorably with an airplane with fixed wings, when it comes to carrying a given load a given distance with given power *in a given time.*

Handley Page has been developing, for a number of years, "slots" for wings. The auxiliary entering edge vanes and trailing edges of wings to form these slots may be opened and closed during flight. The purpose is to delay discontinuity of air flow above an airfoil, or "burbling," with resultant loss of lift, until a very high angle of attack has been reached. The result is that a given airplane, with given wing area, can land at a given speed with about sixty per cent more gross weight (perhaps over a hundred per cent more pay load) than it could without the slots.

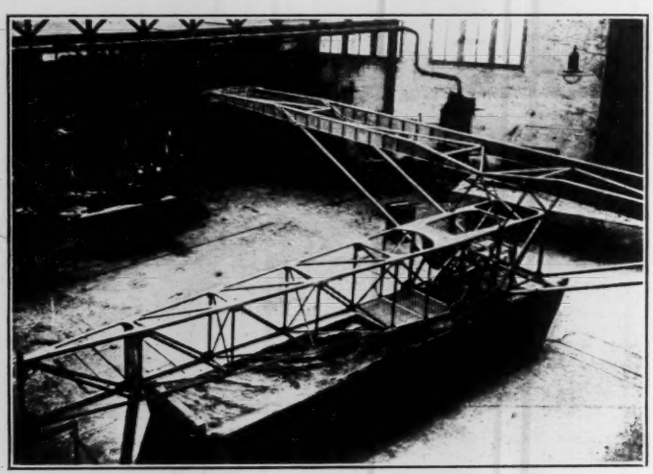

FIG. 4 ALL-METAL STRUCTURE OF WIBAULT MONOPLANE

FRANCE

France has built a tremendous number of military airplanes. Tourists in Europe may hold that the experienced constructors of France have built so many military machines that they can never learn to build *comfortable* passenger transports.

French airplane ideas are well diversified. Many of their modern machines are "sesquiplans"—with a wing and a half—the upper plane having much greater spread, and usually greater chord, than the lower. The famous Breguet XIX, which has flown 3400 miles cross-country without landing, exemplifies this type. There are also many monoplanes braced externally by a single pair of tension-compression struts extending diagonally out from the bottom of the fuselage. This very attractive type was developed by Loening in the United States and by the Morane-Saulnier Company in France. The Wibault is a very interesting French machine of this type—an all-metal monoplane of rather simple construction covered by thin sheet "Alferium." In his latest machines Wibault uses no wires, all loads being taken by compression-tension members.

French engineers seem to have no fear of the "rack and ruin" from landings. This is evidenced in three ways: (1) The very small travel of shock absorbers; (2) The tendency to attach important structural members for wing bracing to the undercarriage; and (3) The general lightness, even flimsiness, of the structure throughout the airplane—ill-suited, one would think, to withstand the racking incident to hundreds of landings—average landings as we know them in this country. It may be that all French pilots are artists at landing, or that they are expected to land only on fine fields, or it may be that French engineers have learned how to make structures very flexible and shock-absorbing and yet strong and durable. Or perhaps the military are willing to sacrifice an airplane every so often in order to save weight and the last word in performance. At any rate the French school in this respect is far different from ours.

The Société d'Emboutissage et de Constructions Mécaniques has installed a large equipment of special machinery toward reduction to real production methods. Their construction is largely of stamped sheet metal in shapes and combinations which look expensive but may not be.

GERMANY

There is no more interesting development than that in Germany. Restrictions imposed by the Allies after the armistice made it illegal for Germany to build airplanes which even remotely resembled military machines. Certain other restrictions at that time put a premium upon getting the most out of a limited amount of horsepower. These rules have forced German engineers into lines of endeavor which may result, within the near future, in Germany's leading the world in commercial aviation. Their airplanes are, from the time of first conception, designed with the sole thought of commercial utility. Except in a few instances, the general design schemes of other so-called commercial airplanes are modifications of those of previous military machines. Furthermore, German engineers and scientists have been forced into a deal of fundamental research, at which there are none more thorough and systematic, in aerodynamics and in the development and use of suitable materials.

The most successful engineers today, Dornier, Rohrbach, Junkers, and others, apparently agree on at least two important principles, viz., (*a*) the monoplane, and (*b*) the stressed skin or covering of thin duralumin sheet.

Junkers has always used his skin, on wings, fuselage, and other surfaces in the corrugated form, which has a distinct advantage in resisting "drum head" vibration and resulting fatigue failure at rivet-attachment points. Corrugated sheet has of course the added advantage of being capable of taking compression loads in, and bending loads normal to, the direction of the corrugations. It has, however, no rigidity in the other direction, is hard to work in such manner as to give a neat appearance where it is necessary to change to a "flat" along edges, and there is some question as to the efficiency of rivets as they must be pitched in a curved surface. Junkers has preferred the straight cantilever wing, strongly tapered both in plan and depth, usually located at the bottom of the fuselage with considerable dihedral. In this cantilever he has not used prime beams but, rather, has replaced chord members by a number of round seamless duralumin tubes, distributed throughout the entire wing section and triangulated together by riveted duralumin strips, beaded to give rigidity in compression except at the ends which are flat and riveted to the tubes. These strips lie in every direction necessary to complete, in conjunction with the skin, the trusses for lift, drag, and torsion.

The structure is very rigid but, to all appearances, very costly in man-hours, because of the tremendous number of rivets which must be headed from the inside of tubes and at other inaccessible or, at best, uncomfortable places.

Notwithstanding the fact that a closed tube is the most economical (from the strength-weight standpoint), reliable, and determinate section with which to take compression loads, and avoids *edges*, which, in duralumin, give us lots of worry, it is a bad thing to which to rivet, as the rivet heads inside cannot be inspected.

Dornier, who builds flying boats and land-type airplanes, convertible as seaplanes, locates his wing above the fuselage, usually with some gap between. His wing is not tapered either in plan or depth. It is braced externally by long struts attached to the bottom of the fuselage, one leading to each of two main wing beams. These reduce by half the length of cantilever. For normal flying loads, these struts are in tension. Being long columns, they are weaker in compression to take reverse loads, but the reverse loads are of far less magnitude than the normal loads. His two wing beams and other important highly stressed parts are of formed heat-treated alloy steel. He uses flat thin-sheet duralumin skin to take stresses from drag and other loads. The skin is stiffened by channels or flanged U-sections, laid at right angles to each other. The channels

lying in direction of flight are riveted outside the skin, top and bottom, and take the place of the compression cap strips in our old-fashioned ribs. Thwartship channels are riveted inside the skin. His airplanes are covered all over by duralumin sheets, except the ailerons, where he uses fabric to reduce weight and static hinge moment, thereby reducing the danger of wing flutter.

On his flying boats lateral stability on the water is obtained (?) by streamline sponsons built out from the main hull. On the Super-Wal two engines are mounted in tandem above the center of the wing.

Both the Super-Wal and Rohrbach's latest flying boat have succeeded in getting off the water with a useful load greater than the weight of the empty machine with power plant.

Rohrbach, who specializes in rather large flying boats, has evidently given a great deal of attention to the reduction of production costs. This has lead him toward an adaptation of shipbuilding practice.

His monoplane wing is cantilever except that, by means of two tightly strung wires leading down and inboard from a station about half-way out, he imposes an initial bending moment and stresses on the inboard part of his wing girders of the opposite nature to those imposed in normal flight, thereby reducing the latter. In spite of the fact that the wing is well above the hull, he has chosen to use a very strong dihedral. At first glance one would say that the resulting lateral stability would be so great that maneuverability would be poor. He has stated, however, surprisingly enough, that this dihedral is necessary *to reduce instability*, there by, of course,

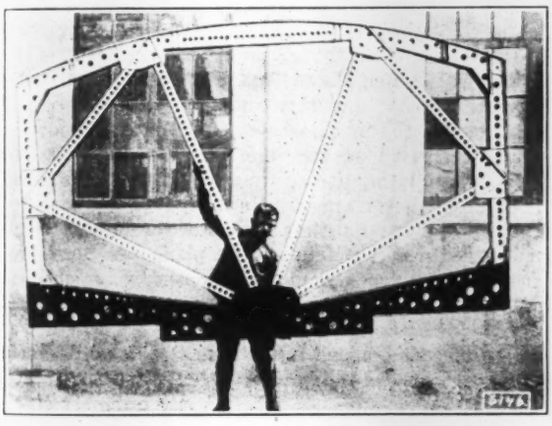

FIG. 5 DURALUMIN BULKHEAD OF DORNIER SUPER-WAL SEAPLANE

improving maneuverability. Until very recently his wing has been without taper in plan, but now, we hear, he uses some taper. His wing loading is extraordinarily high.

His study toward reducing production costs has been along the lines of dividing the wings and other major parts into a great number of separate units. He finishes fabrication of each of these, which permits a great number of men to work without getting in each other's way, and makes the final assembly very simple. Comparatively unskilled labor may be used.

For example, instead of building two wing beams, attaching a lot of ribs, and then applying a covering on this skeleton, he does something like this: The wing is divided into three parts by planes perpendicular to the chord and extending athwartship. The prime structural part is a sort of box girder and includes what we ordinarily call our two wing spars, together with drag bracing, and everything that is located between the two spars. The rounded entering-edge and the trailing-edge sections are built up separately and bolted in place in final assembly. These are readily removable for purposes of inspection and repair. All parts of the wing are of flat sheet metal, with angle sections for mutual attachment. The prime structural box girder is like a long thin caisson or tank, made up, following his principle, of a number of sections, covered top and bottom with comparatively thick imperforate duralumin sheet which serves to resist drag loads. At front and rear are his "wing spars," each made up of a web member and chord members. The web member is of heavy sheet with lightening holes which provide access to the inside of the "tank" for riveting and inspection. The chord members of the individual spars are double angles to which the skin, top or bottom, is attached. Between the four walls of the "tank," extending fore and aft, are a number of fore-and-aft "baffle-plate bulkheads" each of which takes the place of the web of the central portion of an old-fashioned rib. These also are of flat sheet with lightening holes. All these parts are built in sectional units and then the entire box girder riveted together by angle and channel pieces.

HOLLAND

The Fokker Company has adhered to the practices which it originated and which it has found successful through many years. The wings are cantilevers, tapered in plan form and depth from root to tip. The cantilever principle is used in biplanes as well as monoplanes. The wings are entirely of wood, the two main spars being a box section with laminated-spruce chord members and

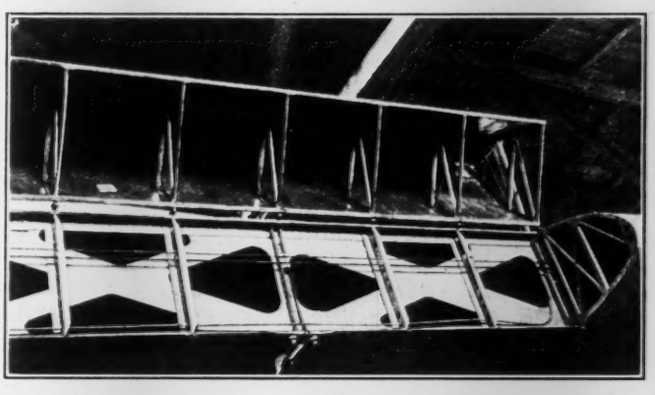

FIG. 6 HINGED LEADING EDGE OF ROHRBACH WING SHOWING ACCESSIBILITY OF ALL PARTS OF WING

double plywood side webs, with vertical spruce members placed vertically within the box at proper intervals, and all glued, screwed, and tacked together. The ribs and covering are also of plywood, the covering taking all drag loads and eliminating the necessity for internal drag trussing. The structure of fuselages is entirely of welded seamless steel tube, the forward portion usually in the form of a modified Warren truss and the rear portion a Pratt truss, with diagonal wires as tension members. Fokker machines have always been noted for their ease of control and stability.

UNITED STATES

Briefly, in our country we build military biplanes, usually externally braced, sesquiplanes in our single-seater pursuits, and many monoplanes for commercial uses. Our fuselage structures are tubular, usually welded seamless steel, but sometimes duralumin. Our wing spars are usually of spruce, and our covering of doped cotton.

The particular phase of aviation in which we, without argument, stand at the head of the world is our Air Mail Service, which operates at night over bad country and in all sorts of thick weather. Our Air Mail Service has been, in this work, remarkably successful, largely due to the pilots and to the efficiency of ground maintenance service. However, one cannot help believing that, even with the installation of light beacons, emergency landing fields, etc., our air-mail pilots, in their efforts to keep up their fine record for reliability, are still operating under conditions which are prohibitively hazardous. A lot must be done yet toward protecting them. It has been suggested that, for night flying in thick weather across bad country, a string of "Neon" tube lights, spaced at intervals of about 300 yards along the entire course, might go far toward solving the problem, but one wonders, when contemplating the cost of light beacons, landing fields, wrecked airplanes, radio installations, meterological service, etc., whether there may not be some other way more reliable, less costly, and *less hazardous to human life* than our present transcontinental Air Mail Service, which is getting more and more complicated as we learn more about it.

The airplane has the peculiar great inherent advantage of being able to travel swiftly at any time in a straight line from any one place

to any other. But when we have a set route such as the New York-Chicago–San Francisco line, perhaps some such means as a streamline container, with no wings, riding on an overhead cable, with or without motorman, propelled by electric motor or by airplane propeller-motor unit, might be better, all things considered.

POWER PLANT

There has been, since the war, great development of, and an encouraging degree of confidence in, air-cooled engines. The most popular of these are of the fixed radial arrangement, usually nine-cylinder, one piston having a master connecting rod, with the other eight rods connected thereto, all to one crankpin. The Bristol "Jupiter," developed in England and built in large numbers, under license, in France, and the Pratt & Whitney "Wasp" in the United States, have proved that 450 hp. is practical in a unit. For this horsepower the radial air-cooled engine appears to be the lightest type, when means of cooling is included. How much more power it is good to take through one crankpin must be proved.

The inverted "V" and "in-line" arrangements, which have certain advantages, have proved practical.

The Napier Lion in England and the Hispano-Suiza "Broad-Arrow" in France are fine examples of "close-coupled" water-cooled engines. Each has twelve cylinders, arranged in three banks of four. Italy, with her "Schneider Cup" Fiat has shown that it is possible to get 900 hp. for 900 lb., exclusive of cooling system.

All these engines are essentially for military use, however. Cost of construction, operation, and maintenance, which must be paramount for commercial uses, have been neglected.

There are those who argue that in using more than one engine we are only admitting weakness in power-plant reliability or in lack of landing fields. This is true, and it may be that the day is almost here when confidence in our power plant (engine proper and ignition, lubrication, fuel supply, and cooling systems) and in having a landing field always fairly handy, will have grown to the point where we shall willingly put all our eggs in one basket. The compromises in the multi-engine arrangement are many. Parasite air resistance increases badly, engine controls and fuel-supply systems become complicated, and often we must have several rudders in order that at least one may always be in the slip stream of the propeller that's turning. There is, of course (except in the case where it is necessary to provide proper fields of vision for a bomber), no excuse for division of power plant unless the airplane can fly with one engine out, and it is very confusing and discouraging to try to calculate a proper, business-like airplane that can really fly, with full load, with one engine out. What we are after, in commercial aviation, is to carry the greatest number of pounds a given distance with the minimum of horsepower. When we reduce the load carried by an amount such that the airplane can really fly with half, or a third, of the power gone (more than that, really, because of reduced propeller efficiency with engines full out at reduced air speed), we are not carrying enough load per horsepower installed to be operating economically.

SUMMARY

As proof that the science is yet fluid, the author cites the following:

England has no welded structures; we use little else in our fuselages.

Fokker developed the welded steel-tube fuselage structure in Germany and still uses it. We in the United States followed him, but Germany did not, nor has any other nation.

France has wooden fuselages and wings of metal structure; we have metal fuselage structures and wooden wing structures.

England uses little duralumin, Germany little else, France some, we less.

England is for biplanes, Germany for monoplanes.

England's biplanes have thin wings with, usually, two-bay cellules. Our biplanes are usually single-bay, with moderately thick wing sections.

Several types of monoplanes are in use: (*a*) The pure cantilever, tapered in plan and in depth (Fokker); (*b*) The cantilever with Rohrbach's wires under initial tension; (*c*) The semi-cantilever, with constant chord and depth, braced underneath by diagonal tension—compression struts (Dornier); (*d*) The semi-cantilever, with diagonal struts, with taper in depth and diagonal or chord from root to tip. There has been a leaning toward type (*c*) in France, Germany, and, lately, in the United States.

Discussion

COLONEL CLARK'S paper was discussed by Messrs. E. E. Aldin,[1] H. C. Pratt,[2] A. H. Lane,[3] O. F. Gashe,[4] Earl D. Osborn,[5] A. F. Denham,[6] and Arthur Nutt.[7]

Replying to the various discussers in turn, the author said that there were at least three different designs of wheel brakes and they were working out very well. They reduced the length of landing run. Also by having a wide tread for the wheels, and by working the brake independently on each wheel, it was possible to secure great turning ability when taxiing on the ground. In a number of machines the rudder control was hooked up with the brake control. Thus by pressing with the left foot it was possible to give left rudder, and at the same time to put the left wheel brake in action. This gave a very powerful control. Braking might conceivably allow a machine to be landed faster with safety. There was a distinct advantage in being able to use machines with high landing speed as a higher maximum speed could then be obtained and a heavier loading per square foot employed. A higher loading per square foot also gave a steadier machine in the air. There certainly was a danger of the machine's turning over when brakes were applied and this was met by putting the center of gravity further forward. The limiting position of the center of gravity relative to the wheel axle was probably such that the angle between the line joining the center of gravity of the airplane and the vertical through the center of gravity did not exceed 40 deg., with the propeller horizontal. In the accident to the Fokker-designed *America*, the full contemplated load had been placed forward of the center of gravity, and this combined with the soft landing ground had turned the machine over. He did not know whether brakes had been used at the time of the accident. Certainly Fokker who was piloting at the time would not have put on the brakes in an injudicious manner. He thought that engine designers were far ahead of plane designers as regarded weight. With regard to reducing the overall weight of the plane per horsepower, he thought that the tendency would be the other way, since business men would want their airplanes to carry as much weight in the form of pay load as possible. He could not say whether aerodynamic design had gone ahead faster than structural design or not. He thought that sheet duralumin was the aircraft material of the future. While welded-steel construction was probably cheaper for small production, up to, say, 200 machines, for larger production he thought that riveted duralumin construction could be so tooled up as to be cheaper. In regard to the proper power for a commercial airplane, he did not yet know what the commercial airplane was, and no definite answer could be given to such a question. In the matter of streamlining, it was always desirable to reduce head resistance to a minimum. He thought that the reason airplanes used during the war were mainly biplanes was that we had at that time understood the structural design of biplanes much better. With the first monoplanes built there had been difficulty with wing flutter while a biplane had great depth between its wings and as a result was very rigid. With monoplanes there was the possibility of carrying a greater useful load per horsepower than with biplanes; again with a monoplane passengers could obtain a better vision downward, as they had no lower wing to interfere with their vision. With monoplanes, however, it might be necessary to use tapered wings, and the cost then went up. This was something to be considered. The German soaring machines were really gliders that worked in an upward current of air, and hence the upward current was indispensable to their long stays aloft. They were undoubtedly good from an aerodynamic standpoint.

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[2] Mechanical Engineer, Rochester Bureau of Municipal Research, Rochester, N. Y.

[3] Chairman, Buffalo Section, A.S.M.E., Buffalo, N. Y.

[4] Hamburg, N. Y.

[5] Vice-President, Gardner Publishing Co., New York, N. Y.

[6] Representative, Automotive Industries, Detroit, Mich.

[7] Chief Motor Engineer, Curtiss Aeroplane & Motor Corp., Buffalo, N. Y.

Electrically Welded Reinforcing Straps on Boilers and Containers

Particulars Regarding Early Investigations Conducted in Switzerland; Contributed by the Author as a Supplement to His Article on Autogenously and Electrically Welded Boilers and Containers, Published in the June, 1926, Issue of This Journal

By E. HÖHN,[1] ZURICH, SWITZERLAND

IN THE June, 1926, issue of MECHANICAL ENGINEERING there appeared an article by the present author entitled Autogenously and Electrically Welded Boilers and Containers. In this various boilers and pressure vessels are shown, some of which are oxyacetylene welded but most of which are electrically welded. These boilers and containers are all in operation in Switzerland at present. However, there are only a few of them, as in Switzerland riveting is still the general practice. We should not have dared to

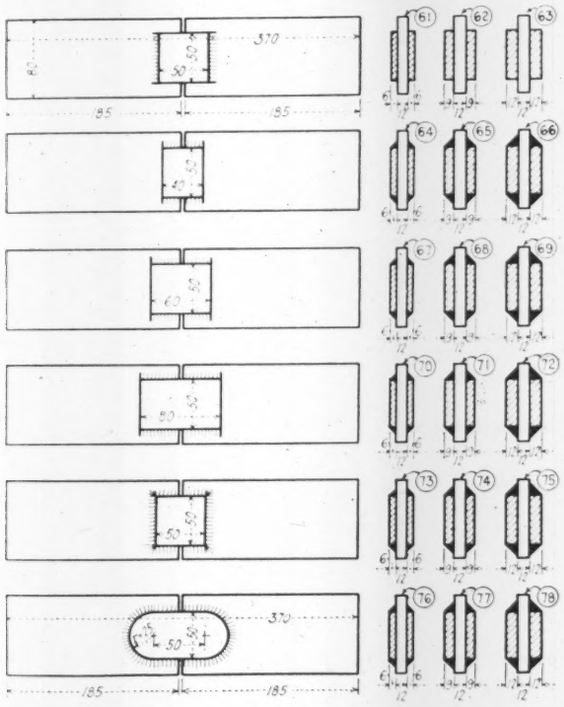


FIG. 1 TEST PLATES WITH DOUBLE STRAPS WELDED ON

weld boilers and pressure vessels electrically had it not been for the insurance against sudden fracture—that is, explosion—afforded by straps across the weld. In Switzerland, it might be noted here, before boilers were welded with the application of straps over the weld for safety, a great many tests on the adherence of such straps to plate were carried out by the author at the Materials Testing Laboratories of the Zurich Institute of Technology. The theory of strap pieces is now cleared up to some extent, and the author would call attention to his articles on The Strength of Electrically Welded Containers, 1923, and Riveting and Welding of Boilers, 1925.

NATURE OF THE TEST PIECE

The work described in the above-mentioned article of 1923 may be summarized as follows:

Electric welding makes it possible to easily weld straps to plate. Autogenous welding does not lend itself to this. The question arises whether such straps have sufficient adherence in order to serve as reliable reinforcement. The author has endeavored to solve the problem by tests on pieces on which double straps are welded (pieces 61–78). Information as to their shape and size may be had from Fig. 1. Pieces 61 to 78 are thin (0.48-in. plate), while (not shown in Fig. 1) 161–178 are thicker (0.67-in. plate). For a strap an ordinary rectangular or rounded piece of plate will serve. It may be welded at the two ends (61 to 63) or at the sides (64 to 72). End welding is subjected to tension and shear, but, in contrast to a lap joint, there is no bending. The adherence strength (per sq. in. end surface) δ is a function of the area of end surface and not of the fractured surface, which is not the same as the end surface. If F is the area of one end surface (of four), then each side of the test piece has two end surfaces. For 61 to 63,

$$\delta = \frac{Q}{2F}$$

The welding material on the sides, 64 to 72, is subjected to shear. The shear strength (rupture load for shear per sq. in. of side surface) is

$$\gamma = \frac{Q}{2F'}$$

Of the total side surface $4F'$, half lies on each side of the test piece, and in the case of 64 to 72 must take the entire load. Pieces 73 to 78 are welded all around. Here end welds and side welds work together. However, δ and γ are determined by the respective sizes of the welds. In all, 108 pieces were tested.

In the tables, E, H, and G as well as J, U, and W designate test pieces from various firms, namely, six. The values given for E, G, and H are an average of the three.

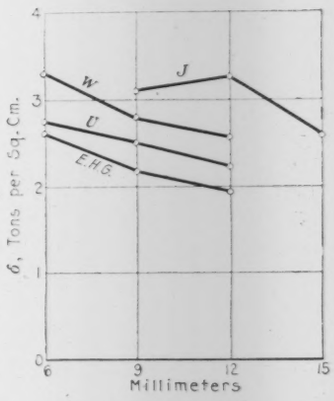


FIG. 2 ADHERENCE STRENGTH δ OF THE END WELDS AS A FUNCTION OF THE STRAP THICKNESS
(Points correspond to Table 1.)

TABLE 1 ADHERENCE STRENGTH OF END WELDS FOR STRAPS 0.24 IN. TO 0.59 IN. THICK

Series	Profile measure	0.24-in. Strap, Pattern 61			0.35-in. Strap, Pattern 62–161			0.48-in. Strap, Pattern 63–162			0.59-in. Strap, Pattern 163		
		2 F sq. in.	Q 1000 lb.	δ 1000 lb. per sq. in.	2 F sq. in.	Q 1000 lb.	δ 1000 lb. per sq. in.	2 F sq. in.	Q 1000 lb.	δ 1000 lb. per sq. in.	2 F sq. in.	Q 1000 lb.	δ 1000 lb. per sq. in.
E H G	21	0.97	35.8	36.9	1.45	44.5	30.6	1.71	48.2	28.2	..	..	..
J	21	..	..	..	1.52	66.6	43.6	1.68	78.1	46.4	19.9	72.4	36.4
U	22	1.02	39.4	38.6	1.43	50.6	35.2	1.90	59.2	31.2	..	..	..
W	23	1.05	48.6	46.4	1.47	57.8	39.3	1.83	65.5	36.0	..	..	..
Max.		..	..	46.4	..	..	43.6	..	..	46.4	..	..	..
Min.		..	..	32.5	..	..	29.6	..	..	24.4	..	..	..

TABLE 2 AVERAGE VALUES

Plate thickness, in.	δ 1000 lb. per sq. in.
0.24	32.5 to 46.4
0.35	29.6 to 43.6
0.48	24.4 to 46.4
0.59	19.7[1] to 36.4

[1] Extrapolated.

It should be stated here that the halves of the test pieces were not welded in the joints in order that the fracture would take place entirely in the straps.

END WELDING

For the test pieces W the profile of the weld was convex, that is, raised. For U it was concave, that is, sunken or hollow. The two series were performed by various welders, but otherwise under

[1] Chief Engineer, Swiss Association of Steam-Boiler Owners.

equal conditions. The purpose was to find out whether a weak profile U was sufficient. In Table 1 it is obvious that W is much stronger than U. From this we may conclude that the profile of an end- or side-weld seam should be convex for straps on boilers and pressure vessels.

The maximum and minimum values are given in Table 1. Values in the table are graphically shown in Fig. 2. It follows from these that the specific adherence strength δ decreases with increased strap thickness.

SIDE WELDING

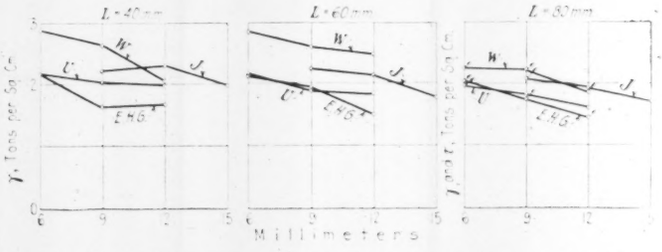
The results of tests on pieces welded only on the sides are shown in Table 3. In such cases it should be noted that the fracture takes place either in the weld, that is, the straps are torn from the plate (indicated by N in Table 3), or where the welds are stronger than the straps, the straps themselves break in the center (indicated by L). In the latter case the weld is stronger than shown in the table (indicated by >).

The values in Table 3 are shown graphically in Figs. 3, 4, and 5. Points designated with a circle show shearing strength. Points designated by > in the table are accompanied by an arrow in the diagram. At rupture, curves with the arrow heads lie higher. It is seen from the table and curves that the shear strength γ of the side welds decreases with increasing strap thickness. The strength γ of the side weld is naturally less than the adherence strength δ of the end weld, because the side weld is subjected only to shear and is a function of the shear strength. On comparing the various tables we see that the relation of γ to δ is 80 per cent for 0.24-in., 73 per cent for 0.35-in., and 68 per cent for 0.48-in. straps.

In the case of straps 1.57 in. and 2.36 in. in length, rupture takes place in the welds. In the case of those 3.14 in. long some of the rupture takes place in the welds, and some in the straps or in the plates. In the case of failure in the weld or 2.36-in. straps the straps were in most cases already cracked. From this we can draw the conclusion that when side-welded straps are more than 3.14 in. long, for example, 3.94 in., rupture will take place in the strap with 1.97 in. strap width. With a strap width greater than 0.48 in., one must of course choose straps of other lengths.

The resistance of the end weld, $2F\delta$, and that of the side weld, $2F\gamma$, are here added. In the case of square straps $2F = 2F'$. If we calculate the stress $Q = 2(F\delta + F'\gamma)$ at which square plates break in the weld on the basis of the minimum values for δ and γ we obtain for the 0.24-in. straps with $F = F' = 1.97 \times 0.24 = 0.45$ sq. in., $Q = 52{,}100$ lb., and for the 0.35-in., 0.48-in., and 0.59-in. straps, 68,500 lb., 79,500 lb., and 85,800 lb., respectively.

These calculations are sometimes the same as, but at other times less than, the figures given in Table 5 from 73 to 75 and 175. We can conclude from this that square straps, when the welds are only of average weight, may break in the seams. If a strap is to break in the center before being torn off, the sides must be longer than in the case of a square. For 1.97 in. strap width the shortest length allowable is 3.14 in. with a strap thickness of 0.48 in. Greater lengths should be allowed for thicker straps.



FIGS. 3–5 SHEAR STRENGTH AND SHEAR REQUIREMENTS OF THE SIDE WELDS AS FUNCTIONS OF STRAP THICKNESS AND LENGTH
(Arrows indicate values of τ; points correspond to Table 3.)

TABLE 3 SHEAR STRENGTH OF SIDE WELDS FOR STRAPS 0.24 IN. TO 0.59 IN. THICK AND 1.57 IN. TO 3.14 IN. LONG
(W indicates fracture in weld; L, fracture in strap)

	$2F'$ sq. in.	Q 1000 lb.	γ 1000 lb. per sq. in.	Break	$2F'$ sq. in.	Q 1000 lb.	γ 1000 lb. per sq. in.	Break	$2F'$ sq. in.	Q 1000 lb.	γ 1000 lb. per sq. in.	Break	$2F'$ sq. in.	Q 1000 lb.	γ 1000 lb. per sq. in.	Break
Length 1.57 in.	0.24-in. strap, No. 24				0.35-in. strap, Nos. 65–164				0.48-in. strap, Nos. 66–165				0.59-in. strap No. 166			
E H G	0.78	23.7	33	N	1.16	26.1	22.6	N	1.42	33.1	23.3	N	..	..	..	..
J	..	..	..	...	1.12	34.5	30.5	N	1.44	45.4	31.6	N	1.75	48.3	27.5	N
U	0.83	24.9	30	N	1.16	33.1	28.4	N	1.48	38.7	26.1	N	..	..	..	..
W	0.80	32.2	40.3	N	1.15	37.3	28.4	...	1.47	41.6	28.3	N	..	..	..	..
Max.	..	..	40.3	...	..	..	28.4	...	..	..	31.6	...	..	..	..	..
Min.	..	..	23.4	...	..	..	22.4	...	..	..	19.3	...	..	..	..	..
Length 2.36 in.	0.24-in. strap, No. 67				0.35-in. strap, Nos. 68–167				0.48-in. strap, Nos. 69–168				0.59-in. strap, No. 169			
E H G	1.17	34.5	29.5	N	1.75	46.1	26.2	N	2.15	44.6	20.7	N	..	..	..	..
J	..	..	..	...	1.85	57.6	31.2	N	2.15	63.0	29.3	N	2.71	66.6	24.6	N
U	1.16	34.6	29.9	N	1.75	45.6	26.2	N	2.16	55.2	25.6	N	..	..	..	..
W	1.14	45.4	39.9	N	1.69	52.7	31.2	N	2.16	63.4	29.5	N	..	..	..	..
Max.	..	..	39.9	...	..	..	31.3	...	..	..	29.5	...	..	..	..	..
Min.	..	..	23.4	...	..	..	22.4	...	..	..	18.7	...	..	..	..	..
Length 3.14 in.	0.24-in. strap, No. 70				0.35-in. strap, Nos. 71–170				0.48-in. strap, Nos. 72–171				0.59-in. strap, No. 172			
E H G	1.64	44.7	>28.5	2L	2.34	58.4	>24.8	2L	2.85	58.3	>24.1	L	..	..	..	..
J	..	..	..	...	2.46	71.5	>28.9	L	2.82	76.7	>27.0	N	3.64	87.9	>24.2	N
U	1.82	49.5	>26.1	L	2.29	57.9	25.3	N	3.00	67.9	22.6	N	..	..	..	..
W	1.75	54.5	>31.5	L	2.22	68.0	>30.5	L	2.96	77.5	>26.1	S	..	..	..	..
Max.	..	..	>31.5	...	..	..	>30.5	...	..	..	27	...	..	..	..	..
Min.	..	..	23.8	...	..	..	23.6	...	..	..	18.7	...	..	..	..	..

TABLE 4 SHEAR STRESS γ, 1000 LB. PER SQ. IN.
(Summary of Table 3)

Strap thickness, in.	Strap length 1.57 in.	2.36 in.	3.14 in.
0.24	23.4 to 40.3	23.4 to 39.9	23.8 to >31.5
0.35	22.4 to 28.4	22.4 to 31.3	23.6 to >30.5
0.48	19.3 to 31.6	18.7 to 29.5	18.7 to 27
0.59[1]	16.3 to 27.5	16.9 to 24.6	16.9 to 24.2

[1] Lower values extrapolated.

STRAPS WELDED ALL AROUND

These are the test pieces Nos. 73 to 78. For 73 to 75 the straps are square, although it was known before that with oblong straps the adherence strength would be greater because of the longer side welds. We purposely tried to tear the square straps from the plate in order to get their adherence strength. However, we were not able to do so, the straps breaking through the center. The oblong straps 76 to 78 also broke. This was even more to be expected than in the case of the square ones.

TABLE 5 TEST DATA ON WELDED JOINTS WITH STRAPS WELDED ALL AROUND
(L indicates break in strap; S, in plate)

	$2F + 2F'$ sq. in.	Q 1000 lb.	$\delta + \gamma$ 1000 lb. per sq. in.	Break	$2F + 2F'$ sq. in.	Q 1000 lb.	$\delta + \gamma$ 1000 lb. per sq. in.	Break	$2F + 2F'$ sq. in.	Q 1000 lb.	$\delta + \gamma$ 1000 lb. per sq. in.	Break	$2F + 2F'$ sq. in.	Q 1000 lb.	$\delta + \gamma$ 1000 lb. per sq. in.	Break
	0.24-in. strap, No. 73				0.35-in. strap, Nos. 74–173				0.48-in. strap, Nos. 75–174				0.59-in. strap, No. 175			
E H G	2.08	52.3	>25.2	3L	3.06	70.2	>22.8	2L1S	3.72	>73.9	19.9	1L2S	..	..	..	..
J	..	..	..	...	3.25	89.0	>27.3	L	3.90	>98.7	25.4	L	47.1	107	>22.7	L
U	2.27	64.0	>28.1	L	2.92	69.4	>23.8	L	3.90	>76.2	19.4	S	..	..	..	..
W	2.10	52.1	>28.2	L	3.04	73.8	>24.1	S	3.73	>79.6	21.4	S	..	..	..	..
E H G	2.64	51.6	>19.4	3L	3.93	73.5	>18.7	3L	4.71	>76.0	16.2	3S	..	..	..	..
J	..	..	..	L	3.90	87.9	>22.5	L	4.93	>10.3	20.9	L	5.91	105	17.9	S
U	2.79	57.4	>20.6	L	3.93	76.8	>19.6	L	4.93	>70.2	14.2	S	..	..	..	..
W	2.79	61.0	>21.8	...	3.90	67.2	>12.2	S	5.09	>76.7	15.1	S	..	..	..	..

The tests yielded extremely desirable results, as all the failures occurred across the straps and the latter were not torn from the plate proper. When the cross-section through the strap was greater than that of the plate to which it was attached, failure occurred in the latter. The condition of the weld in the case of failure in the strap or plate is given in Table 5. The letter L denotes a rupture in the strap; S, one in the plate. The condition is calculated as follows:

It is pointed out that the straps do not break in the weld but across, so that we do not know the rupture strength from Table 5 but rather the requirements. That a higher stress is allowable is shown by the symbol >. The requirements in the case of the square straps reach from 28,100 to 19,500 lb. per sq. in. (Table 5,

top) as they approach the rupture strength. With long straps, they are lower, 22,600 to 14,200 lb. per sq. in., so that in the case of the long straps (Table 5, bottom) $L/B = 2$, the safety factor against tearing from the plate, is very great. The sketch of the etched section in Fig. 6 shows that the weld material extends well into the plate material to which the strap is welded. This is the reason for the adherence strength.

SUMMARY OF THE 1925 EXPERIMENTS

The above-described experiments of 1923 were checked in 1925 with heavier plates and straps. The nature and dimensions of the test pieces are given in Fig. 7, each strap being 2.36 in. × 0.79 in. in cross-section. Both straps had a nominal combined cross-sectional area of 3.74 sq. in. (more exactly, 3.64 sq. in.). The sides of the end and side welds were 0.59 in. wide, the length of the sides 3.94 in. for F_1, 5.9 in. for F_2, and 7.9 in. for F_3. The thickness of the strap was chosen greater (0.79 in.) than the height and the side of the weld, in order to cause the strap to be torn from the plate. This should allow us to determine the adherence strength. Test pieces from the firms O and E were welded at the same time, being run in duplicate. For welding, wound electrodes were used. Table 6 gives the results. Here it is shown as in previous tables that the strength cannot be based on the actual fractured surface—which cannot be determined by the designer—but must be based on the end and side surfaces. From the table it is obvious that in spite of its greater thickness, in most cases the strap did not break but failure occurred across the strap.

FIG. 6 SKETCH OF ETCHED SECTION OF DOUBLE STRAP WELDED TO PLATE

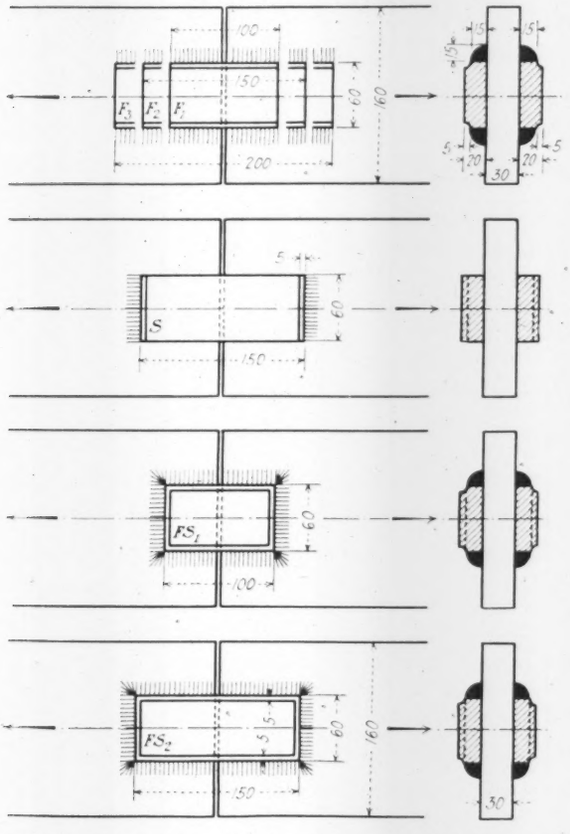


FIG. 7 DIMENSIONS OF VARIOUS TEST PLATES WITH DOUBLE STRAPS—1925 TESTS

A further matter to be determined was that of the elastic properties and toughness present in the strap joints considered. For this purpose the plates were loaded and the widening of the joint between them measured. For measuring, Zeiss dials were used. The total widening of the joint is designated by 2ξ, the permanent elongation by $2\xi'$; for one-half of the test piece we have then ξ as well as ξ'. The results are shown in Figs. 8, 9, and 10 as well as in Table 7. (These refer to series O.) The load is not only given an absolute value in pounds, but for the test pieces S and F comparative values in pounds per square inch are also given, that is, referred to the load per square inch of the end and side surfaces.

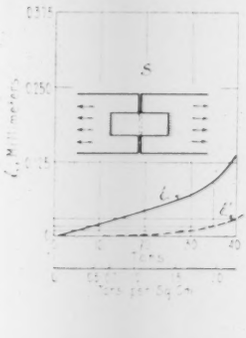


FIG. 8

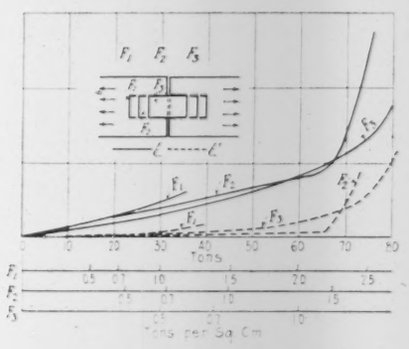


FIG. 9

FIG. 8 TOTAL AND PERMANENT WIDENING (ξ AND ξ') OF JOINT FOR A HALF PLATE END WELDED TO S

FIG. 9 TOTAL AND PERMANENT WIDENING (ξ AND ξ') OF JOINT FOR HALF PLATE WITH SIDE WELDS, F_1, F_2, AND F_3

The last-named measurements are displaced, which indicates that with a specific load the elongation increases with increasing side length. Despite this fact curves F, Fig. 9, are practically identical for the same absolute loads. The reason for this is that the portion of the weld next to the joint stretched practically the same whether the sides were long or short; in other words, a section of the weld further from the joint will remain intact even though the strap in the vicinity of the joint is stretched. The shear stress decreases with increase of the side weld proceeding from the joint. The use of

TABLE 6 ADHERENCE STRENGTH OF TEST PLATES WITH DOUBLE STRAPS

(Length of weld edge, 1.5 cm.)

	Series O			Series E		
ADHERENCE STRENGTH δ IN 1000 LB. PER SQ. IN. OF END WELDS OF STRAPS S	Load, 1000 lb.	δ 1000 per sq. in.	Break,	Load, 1000 lb.	δ 1000 per sq. in.	Break
S (width) = 2.36 in.	110	39.1	Weld	94.6	33.7	Weld
SHEAR STRENGTH γ (1000 LB. PER SQ. IN.) OF SIDE WELDS OF STRAPS F_1, F_2, F_3		γ			γ	
F_1 (half-length) = 1.97 in.	133	29.0	Weld	126	26.9	Weld
F_2 (half-length) = 2.95 in.	186	26.7	Weld	183	>25.9	Strap B = 50.3
F_3 (half-length) = 3.94 in.	220	>23.6	Intact	200	>21.4	Strap B = 55.3
ADHERENCE STRENGTH OF STRAPS WELDED ALL AROUND, FS_1 AND FS_2		$\delta + \gamma$[1]			$\delta + \gamma$[1]	
FS_1 (half-length) = 1.97 in.	220	2.93	Weld	194	>25.9	Strap B = 53.3
FS_2 (half-length) = 2.95 in.	196	?	Head	207	>21.0	Strap B = 57.0

[1] If the value of $\delta + \gamma$ is used only approximate results are obtained, as the value of each should be individually applied.

TABLE 7 TOTAL AND PERMANENT WIDENING OF JOINT (2ξ AND $2\xi'$) WITH TEST PIECES S, F_1, F_2, AND FS_2—SERIES O

(σ_z is the tensile strength of the straps, σ_a of the end surfaces, and T the average tensile strength of the side welds figured on the side surface.)

Straps			S		F_1			F_2			FS_1(4)	
P 1000 lb.	σ_z 1000 lb. per sq. in.	σ_a 1000 lb. per sq. in.	2ξ 0.001 in.	$2\xi'$ 0.001 in.	T 1000 lb. per sq. in.	2ξ 0.001 in.	$2\xi'$ 0.001 in.	T 1000 lb. per sq. in.	2ξ 0.001 in.	$2\xi'$ 0.001 in.	2ξ 0.001 in.	$2\xi'$ 0.001 in.
11	3.03	3.92	0.75	0	2.34	0.47	0	1.56	0.55	0	0.43	0
22	6.05	7.85	1.5	0.08	4.70	1.41	0.08	3.13	1.26	0	1.06	0
33	9.08	11.7	2.3	0.20	7.05	1.88	0.08	4.70	1.88	0	1.65	0
44	12.1	15.7	3.3	0.35	9.40	2.87	0.20	62.5	2.48	0	2.20	0.08
55	15.1	19.6	4.4	0.55	11.7	3.66	0.35	7.85	3.11	0.04	2.75	0.12
66	20.0	23.5	5.6	0.79	14.1	4.65	0.55	9.41	3.82	0.02	3.42	0.20
88	24.2	31.3	10.8	2.22	(2)	..	..	12.5	5.12	0.04	4.80	0.40
110	30.4	(1)	...	...	...	...	...	18.6	6.40	0.08	6.31	0.75
132	36.4	...	...	...	...	...	...	18.8	7.95	0.08	8.35	1.50
154	42.3	...	...	...	...	...	...	21.9	13.6	4.18	11.3	2.87
176	48.4	...	...	...	...	...	...	25.0	12.5	9.68	17.6	7.06
220	60.5	...	...	...	...	...	...	(3)	...	..	(5)	

(1) Fracture in weld at 110,000 lb.; accordingly $\sigma_a = \delta = 39{,}100$ lb. per sq. in. (adherence strength).
(2) Fracture in weld at 133,000 lb.; accordingly $T = \gamma = 39{,}000$ lb. per sq. in. shear strength.
(3) At 220,000 lb. no visible change.
(4) With the test pieces with straps welded all around the specific stress is not shown as it is made up of the unknown values σ_a and τ.
(5) Weld broke at 220,000 lb.; approximate calculation gives $\delta + \gamma = 39{,}400$ lb. per sq. in.

specific stress as a measure of elongation of the side welds is only approximately correct, just as in the case of the average weld cross-section for the calculation of shearing stress when several are acting in combination.

From this we see again that when the strap and the side weld are of sufficient length the use of end welds becomes superfluous. Moreover it is seen in the case of F_2 and F_3 that failure took place in the center, which checks these conclusions.

In the test the load is applied in steps and subsequently diminished to zero. If the needle of the instrument does not return to zero, permanent elongation is indicated. In Figs. 8 to 10 the upper

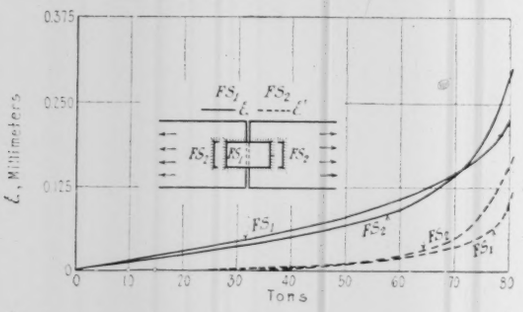


Fig. 10 Total and Permanent Widening (ξ and ξ') of Joint for Half Plate with Straps FS_1 and FS_2 Both End and Side Welded

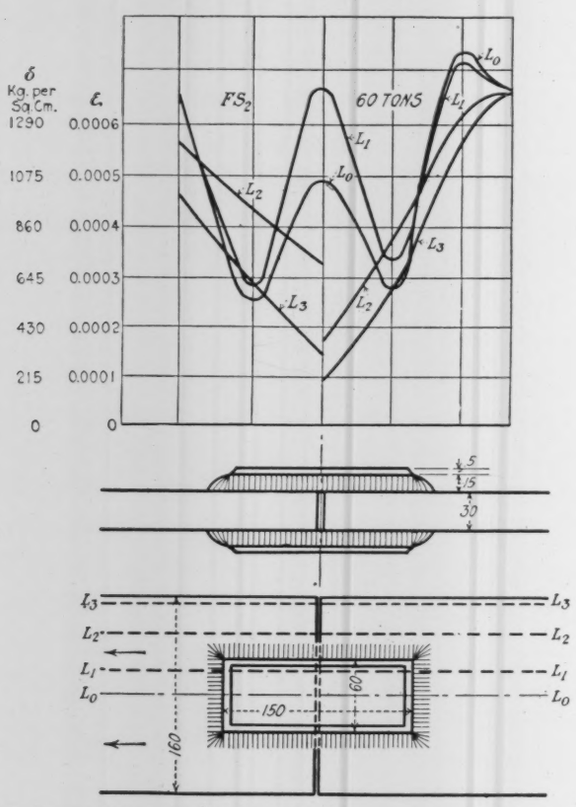


Fig. 11 Stress Conditions in Test Piece FS_2, Series E (Welded at Ends and Sides)

full curves show the total joint widening ξ for one-half of the test piece and the lower broken lines the permanent widening. The joint widens for double the distance between the two curves elastically. In the opening of the joint other factors enter; the elongation of the straps, the elongation of the end weld in the case of test piece *S*, side welds in case of F, and both in the case of FS. In the plate the elongation was small because of the large cross-section. Elongation can be either elastic or permanent. In the simple case of Test S (illustrated by Fig. 8 and Table 7), one can easily see that the total joint widening is greater than the strap elongation according to Hooke's law. Thus the end welds will stretch the others. This much we can know with some degree of certainty, namely, that the strap joint is elastic, and from this the following.

From the condition of the side weld we can deduce that this type of joint also has a certain amount of toughness. This may be deduced from the figures and Table 7. Fig. 11 also gives the stress condition of the straps of FS_2, series E.

The longitudinal stress shows a wave form over the plate and straps. The stress in the halves of the plate drops where the straps begin increasing the cross-section of the material. In the strap the stress increases and reaches a maximum in the center over the joint. This increase proceeds regularly, which is an important point.

Conclusions

Electrically welded straps adhere so tightly to the plate that they themselves will rupture before being torn off, provided the side length is as great as the end length. The adherence strength of a strap is a combination of the adherence strength of the end weld and the shearing strength of the side weld. The end welds are not sufficiently strong of themselves so that a strap will rupture before it is torn from the weld. For this side welds are essential. If the laps in the side welds are long enough so that the adherence strength is sufficient, the straps will rupture in the center without the help of the end weld.

The absolute value of the shearing strength of a good weld up to 0.48 in. height of weld profile is taken as 18,000 to 28,000 lb. per sq. in. The adherence strength of good end welds having the same profile is 24,000 to 41,500 lb. per sq. in. The shearing strength (per sq. in.) of the side weld is less than the adherence strength of end weld (per sq. in.). Both stresses fall off with increased weld profile.

The profile of a good side weld should be as full as possible—certainly not concave—and the lower limit boundary line should be straight.

If straps of the form FS_2, illustrated in Fig. 7, are welded across the seams on boilers and on pressure vessels on both sides, they increase the strength of the weld up to the full cross-section of the straps.

"Mumetal"

THE first commercial application of high-frequency melting in Europe was made by a British firm for the preparation of nickel-iron alloys for submarine cables. The research work pursued in order to find a suitable alloy was thereby much facilitated, and quickly resulted in the perfection of the series of alloys known under the name of "Mumetal"—so called because the Greek letter μ is the symbol of permeability. These alloys have highly desirable characteristics, and by their use the speed of cabling can be increased seven to eightfold.

A typical mumetal alloy has the following composition: nickel, 74.0 per cent; iron, 20.0 per cent; copper, 5.3 per cent; manganese, 0.7 per cent.

An alloy of this composition has a magnetic permeability of 7000. Low hysteresis loss with a very high permeability at low magnetizing forces are the characteristics chiefly required in these metals.

The composition of the mumetal may be varied according to the type of cable which it is proposed to use, and the annealing may be adjusted to give either (*a*) maximum neutral permeability, (*b*) maximum permeability at magnetic saturation, or (*c*) high or low fields for magnetic saturation.

The range of alloys developed for this class of work contain from 75 to about 50 per cent of nickel, with percentages of iron varying from 20 to 25, and certain quantities of copper and manganese, but in every case freedom from carbon is one of the primary essentials, and for this reason the high-frequency furnace has proved the only suitable melting equipment.

In certain types of cable an increase in electrical resistance of the alloys is desirable, and in this case an extra element, which may be tungsten, chromium, silicon, vanadium, titanium, molybdenum, or aluminum, may be added in small quantities.—*The Engineer*, May 13, 1927, p. 521.

Service Factors Governing the Slagging of Boiler-Furnace Refractories[1]

Progress Report of the A.S.M.E. Special Research Committee on Boiler-Furnace Refractories

By RALPH A. SHERMAN[2] AND EDMUND TAYLOR,[3] PITTSBURGH, PA.

ONE OF the most serious causes of failure of refractories in boiler furnaces fired with coal is the erosion by slag from the coal ash. The total expense incident to this and other types of failure has become so great that there has been a marked tendency in the past two years to protect or entirely replace by water- or steam-cooled surfaces the refractories in the areas most seriously affected. Some boilers have no refractories within the furnace.

The factors governing slag erosion may be divided into two classes:

1 Service factors, dependent on the type of coal burned and the method of burning
2 Refractories factors, dependent on the type of clay used and the method of manufacture.

The purpose of this paper is to deal with the first class of factors, discussing their significance and presenting values obtained in the field investigation of refractories service conditions conducted by the U. S. Bureau of Mines in coöperation with the Special Research Committee on Boiler-Furnace Refractories of The American Society of Mechanical Engineers.

The main objective of this investigation is to supply information to the ceramic engineer to aid him in the study of the mechanism of slag erosion, spalling, and other causes of failure, and enable him to designate the proper application of the present refractories, or point to the development of special refractories.

The investigation of the second class of factors is being conducted concurrently by the Bureau of Standards in coöperation with the Bureau of Mines and the A.S.M.E. Committee.

Scope of Investigation

The service factors that are considered to govern slag erosion directly and are being experimentally investigated in the field for various coals and coal-burning equipment are:

1 Refractories temperatures
2 Furnace-gas temperatures
3 Furnace-gas composition
4 Furnace-gas velocities
5 Slag composition
6 Slag quantities.

At this time, values for part or all of these factors have been determined under each of the following sets of conditions:

1 Eastern bituminous coals on underfeed stokers
2 Powdered eastern bituminous coals in furnaces with extended radiant-heat-absorbing surface
3 High-sulphur Pittsburgh coal on underfeed stokers
4 Low-sulphur Pittsburgh coal on underfeed stokers
5 High-sulphur Pittsburgh coal on chain-grate stokers
6 Illinois coal on chain-grate stokers
7 Powdered Illinois coal.

Refractories Temperatures

It is entirely obvious that the temperature of the hot surface of the refractory wall and the gradient through the wall are of importance in slag erosion. If the surface of the brick is somewhat softened, the molten ash deposited on it adheres and combines much more readily than if the surface were hard.

The temperature gradient through the brick determines the depth of absorption of slag, if any, as this will stop at the plane where the temperature is sufficiently low to cause congealing of the slag.

The maximum surface temperatures of refractories measured in the furnaces investigated ranged from 2500 deg. fahr.[4] in the walls of an underfeed-stoker-fired furnace to between 2700 and 2800 deg. in the arch of a furnace burning Illinois coal on a forced-draft chain-grate stoker and in the side wall of a powdered-coal furnace.

These temperatures were well below the softening temperatures of the refractories ordinarily determined, but were above the softening temperatures and frequently above the fluid or free-running temperatures of the coal ash.

Furnace-Gas Temperatures

The temperature of the furnace gases often determines the temperature reached by the walls, although the temperature of the fuel bed, by radiation to the walls, may be the more important determining factor. The gas temperature also governs the temperature, and hence the plastic state, of the ash carried in the gases to be deposited on the walls.

It is frequently found that the temperature of the gases within four to six inches of a furnace wall is lower than that of the gases within the furnace. This may be caused by air infiltration through the wall or past the ledge plate or side tuyeres of the stoker. It is also frequently true that the temperature of the gases next to the wall is lower than that of the wall. In such a case the temperature of the wall is determined by radiation from the higher-temperature fuel bed or the main body of furnace gases.

Gas temperatures of 2750 deg. have been measured in the underfeed-stoker furnaces studied, but normally the maximum temperatures measured did not exceed 2600 deg. The maximum gas temperatures measured in chain-grate-stoker and powdered-coal furnaces were from 2700 to 2800 deg.

The difference in the gas temperatures in underfeed- and chain-grate-stoker furnaces is accounted for by the difference in the type of construction. Fig. 1 shows the typical construction of these two types of furnaces. The chain-grate stoker is set under an arch, making a "dutch oven" with little of the furnace open to radiation to the boiler tubes, while the underfeed stoker is rarely set under an arch and the entire area of the fuel bed and furnace may radiate to the tubes.

Although the flame in a powdered-coal furnace is generally "in sight of" the boiler tubes and frequently of additional heat-absorbing surface in the walls or bottom of the furnace, the temperatures may be relatively high because of a low excess of air or frequently because the zone of intense combustion is confined to a small part of the furnace which may be shielded from radiating to the heat-absorbing surface.

Furnace-Gas Composition

The most important item in the composition of the furnace gases with relation to slagging is the content of reducing gases—carbon monoxide and hydrogen. The most generally accepted explanation of the greater slagging in the presence of these gases is the reduction of the ferric to ferrous iron and the consequent formation of ferrous silicates, which melt at a lower temperature than ferric silicates.

Operation of Underfeed Stoker

Fig. 2 shows the interior of an underfeed-stoker-fired furnace. Coal is fed from the hopper on the outside by rams which push

[1] Published by permission of the Director, U. S. Bureau of Mines. Presented at the Boiler Furnace Refractories symposium, American Ceramic Society, February 15, 1927.

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[3] Assistant Fuel Engineer, Pittsburgh Experiment Station, Bureau of Mines.

[4] Fahrenheit scale used throughout this paper.

it up through the incandescent fuel bed above. Between the retorts into which the coal is fed are the windboxes from which air is supplied under pressure through cast-iron tuyere plates. The coal thus forms a series of peaks and valleys from side to side of the furnace, the secondary air necessary for the combustion of the volatile matter, carbon monoxide, and hydrogen arising from the fuel bed comes through the valleys and through cracks formed by the shrinkage of the caked coal. The bed varies in thickness from 6 to 10 in. above the tuyeres at the front to a maximum of 18 to 36 in. at the thickest part, approximately half-way to the rear of the furnace, and then decreases in thickness to the dump plates or clinker grinders where the ash is discharged into the ashpit. In the latter part of the coal travel the stoker is partly overfeed in operation.

The great depth of the fuel bed is favorable to the presence of reducing gases directly above it, and considerable quantities (12 to 14 per cent of carbon monoxide has been found) occur along the front wall and along the side wall over the thickest part of the bed.

These stokers were originally used principally for coking coals. Hence the higher-grade eastern bituminous coals are generally burned on them, although they have been adapted to the free-burning, such as Illinois coals, with considerable success. Slag erosion is not therefore generally the most serious refractories problem with these stokers, although deep, cup-like erosion occurs on the front wall above the tuyeres in many installations. The reducing content of the gases is generally greater over the tuyeres, although this is not invariably true.

Effect of Slag Adhesion

The adhesion of slag to the walls is the more frequent problem with these stokers owing to the class of fuel burned on them and the relatively moderate temperatures, and ventilated blocks are frequently installed up to the top of the fuel bed on the front, side, and bridge walls to lessen the adhesion by cooling the wall and maintaining an oxidizing atmosphere. Fig. 3 shows the comparative gas temperatures and excess air above the fuel bed of an underfeed stoker, with solid brick walls and with ventilated refractory blocks, burning a Pocahontas coal. The excess air directly inside the wall is much higher with the ventilated blocks, but the effect did not extend beyond approximately 6 in. from the wall. The gas temperatures show the same condition, and temperatures ranged from 1600 to 2000 deg. inside the solid wall and from 400 to 1000 deg. inside the ventilated wall, but at 4 to 6 in. from the wall the temperatures were approximately the same with either installation. The slag adhesion was less during the period of observation with the ventilated blocks, and the general experience of others has been that they are helpful if care is taken to keep the holes free from slag.

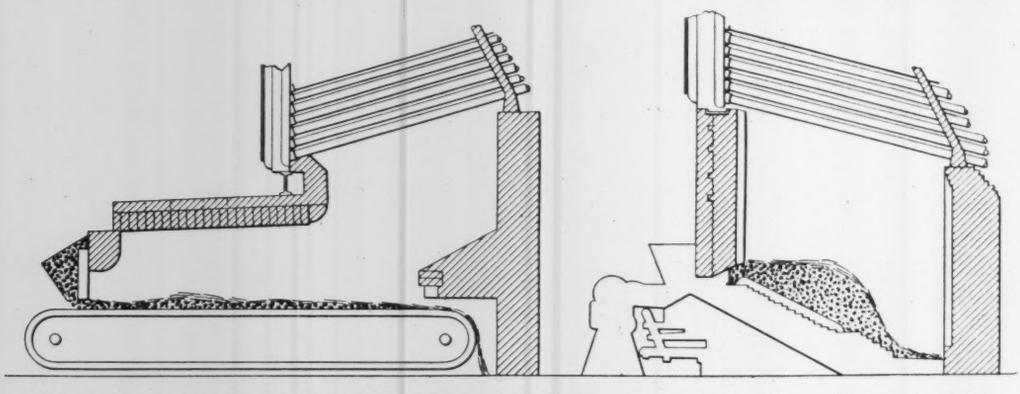

FIG. 1 SECTIONAL ELEVATIONS OF UNDERFEED STOKER (RIGHT) AND CHAIN-GRATE STOKER (LEFT), SHOWING RELATION TO THE BOILER TUBES

Operation of Chain-Grate Stoker

Coal is fed into chain-grate-stoker furnaces from the hopper by the continuous movement of the grate in an even layer, varying from 4 to 7 in., adjusted by the height of the coal gate. The latter figure is not often exceeded in present practice. The coal passes under an ignition arch which serves to reflect heat from the hotter part of the fuel bed on to the green coal. The depth of the fuel bed decreases naturally through the length of the grate. When air is supplied by forced draft, the windbox is generally divided into compartments with dampers so that the air supplied to the distillation and other zones of the grate may be controlled.

There is a zone, usually 2 to 3 ft. from the coal gate—although this may vary with different coals—where the gases arising from the fuel bed contain considerable amounts of reducing gases. These are burned when they meet the excess air coming through the rear part of the bed, or by secondary air admitted at the front

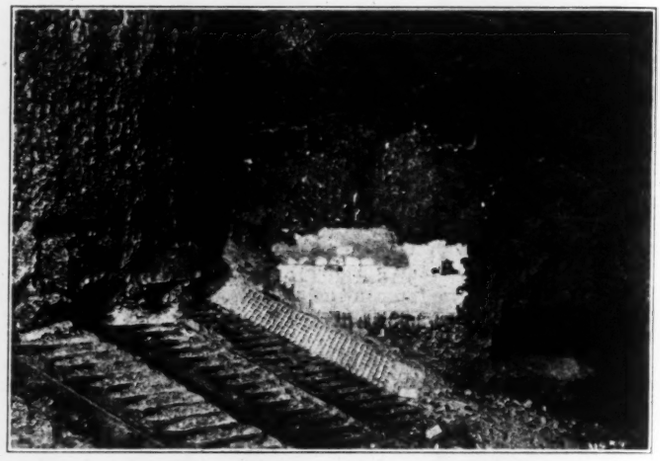

FIG. 2 INTERIOR OF UNDERFEED-STOKER-FIRED FURNACE SHOWING METHOD OF SUPPLYING COAL AND AIR

or through the arch. As the gasification and combustion are rapid in this zone, the gases generally fill the furnace and sweep the side walls.

Fig. 4 shows the side wall of a forced-draft chain-grate-stoker furnace on which Pittsburgh coal had been burned for 367 service hours from the installation of a new wall. There is a deep, pocket-like erosion starting at about 20 in. from the front of the furnace which extends to the back, increasing in width but decreasing in depth. The temperature of the most deeply eroded area was not as high as that of areas which were eroded less, but the gases contained up to 12 per cent carbon monoxide and 5 per cent hydrogen, and were greatest at about 150 per cent of boiler rating.

It was conceived that the admission of air to reach this area might eliminate the reducing gases next to the wall and reduce the erosion. A pipe was installed through the side wall at the nose of the ignition arch which pointed toward the eroded area, and this pipe was connected to the air duct and the air supplied was increased by means of a steam injector. It was found that the reducing gases could be entirely eliminated, and the gas temperatures were reduced from 200 to 300 deg. at the point of sampling. The furnace was operated for a period of about three weeks after building a new wall in this area, with air admitted during all steaming periods.

Fig. 5 shows the wall at the end of this time. The erosion was not entirely eliminated but was reduced in area, as can be clearly seen from the nearly horizontal upper limit of the eroded area.

Operation of Powdered-Coal Furnace

The powdered-coal method of burning differs in principle from those of hand firing or stokers in that all the coal, and consequently all the coal ash, is carried in suspension in the gases. In stoker firing the ash carried in the gases varies with the rate of burning and depends somewhat on the fineness and caking qualities of

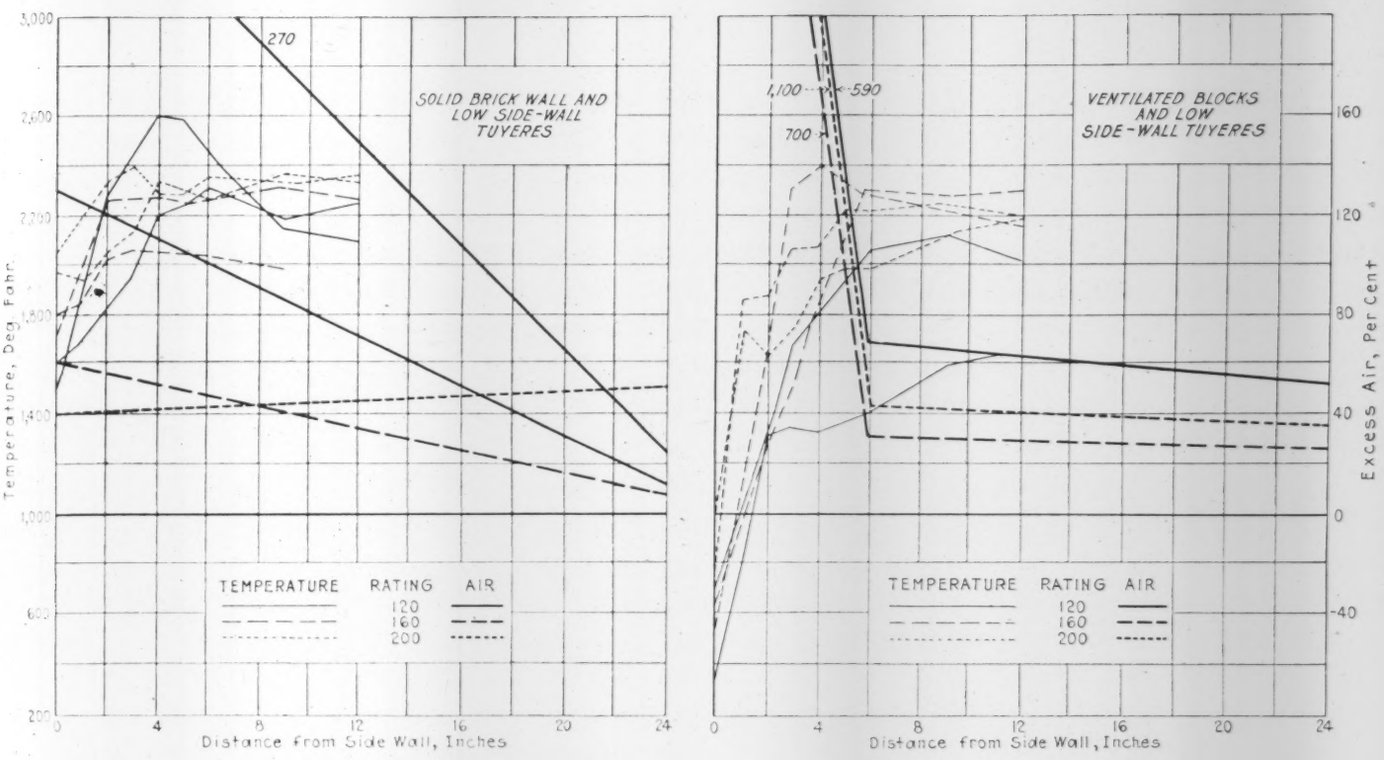

FIG. 3 GAS TEMPERATURE AND CONSTITUTION AT SIDE WALL, UNDERFEED-STOKER FURNACE

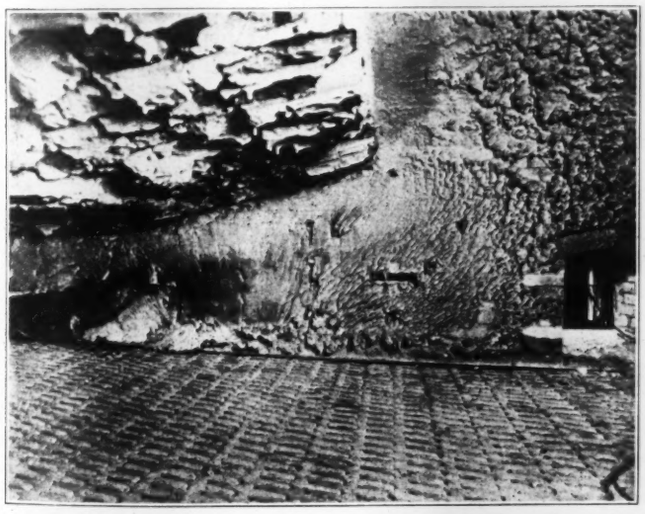

FIG. 4 INTERIOR OF CHAIN-GRATE-STOKER-FIRED FURNACE SHOWING SLAG EROSION ON SIDE WALL

the coal; the maximum amount may be about 20 per cent of the ash in the coal.

The possibility exists that the refractories in a powdered-coal furnace may be bombarded with a greater weight of ash or slag particles, although these may be smaller than those arising from the fuel beds. It is therefore desirable to keep the furnace gases from impinging on to refractory surfaces.

Fig. 6 is a sectional elevation through a powdered-coal furnace in which the coal from a storage bin with about 15 per cent of air (heated by passing through ducts in the ashpit lining) necessary for its combustion is fed vertically through flat burners whose longest dimension is parallel to the side wall; the remainder of the air is supplied by natural draft through ports in the front wall after passing through horizontal lanes in the side wall back of the 9-in. brick lining. The rear wall and the ashpit are protected by water-screen tubes which form part of the boiler heating surface. The side and front walls and burner arch are of refractories.

The flame takes a path downward and then upward in a U as shown by the circles which mark the positions at which gas temperatures and samples were taken. Fig. 7 shows the temperature and composition of the gases at 6 in. and 4 in. from the side wall, respectively, when burning an Illinois coal at three boiler loads of 140, 190, and 240 per cent of rating. The maximum gas temperatures occurred at about 22 ft. from the burners and were 2600 deg. at 140 per cent and 2760 deg. at 190 and 240 per cent of rating.

The CO_2 content was low and the oxygen content high at the first two sampling positions, 6 ft. and 11 ft. from the burners, because the flame did not reach the side wall there, but the CO_2 content increased rapidly and CO and H_2 were found at the three lower positions in the furnace, increasing from 2 to 3 per cent at 140 per cent of rating to 4 per cent at 190 per cent of rating, and to 8 and 12 per cent at 240 per cent of rating.

Fig. 8 shows the side wall of this furnace four weeks after rebuilding the lower part of the wall. There was approximately two inches of erosion in the area above the water-tube slag screen where the gas temperatures and reducing content were at a maximum. Fig. 9 shows a wall which has been carried to complete failure, and shows more clearly the coincidence of the eroded area

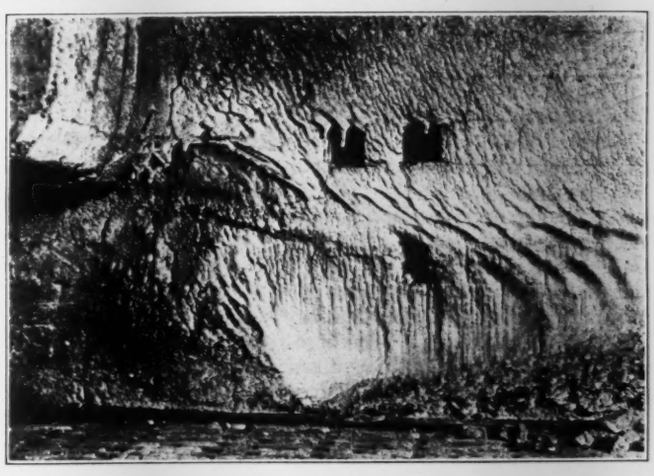

FIG. 5 INTERIOR OF CHAIN-GRATE-STOKER-FIRED FURNACE SHOWING REDUCTION OF SLAG EROSION ON SIDE WALL

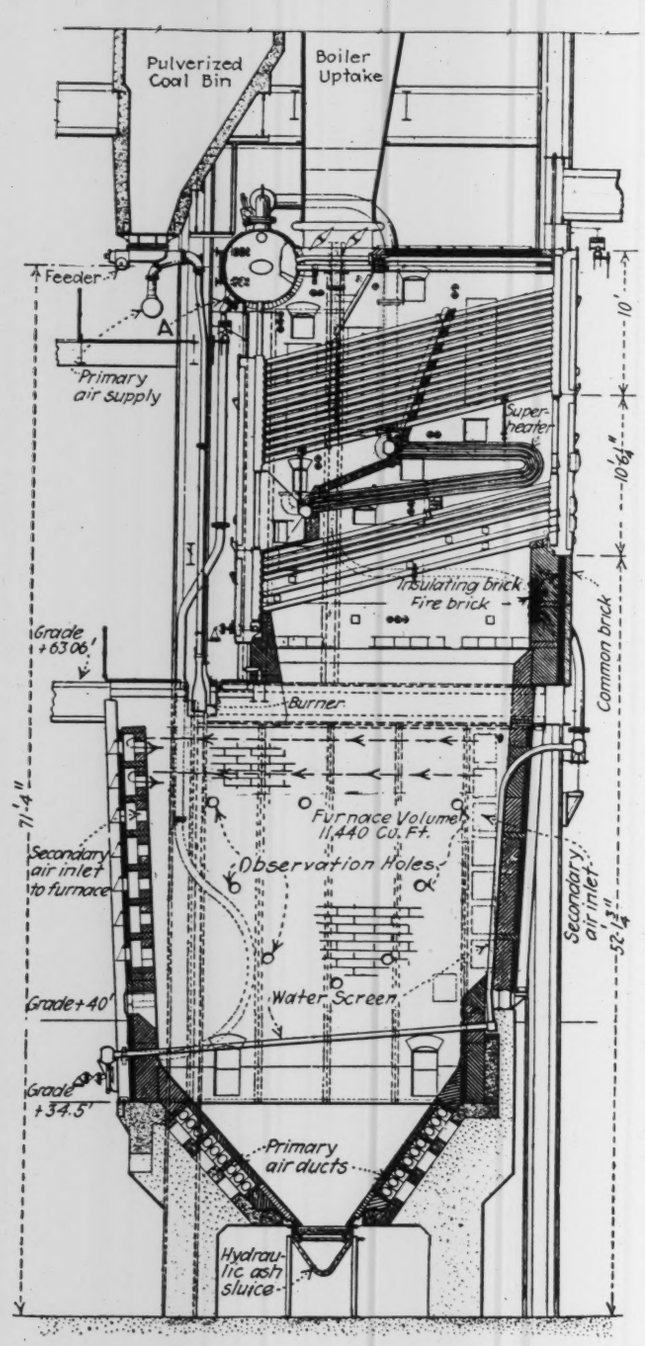

FIG. 6 SECTIONAL ELEVATION OF POWDERED-COAL FURNACE SHOWING POSITIONS FOR SAMPLING GASES AND MEASURING TEMPERATURES

and the area of high temperatures and high reducing-gas content. The deleterious effect of the reducing gases cannot be doubted, and if they were eliminated the erosion would most probably be lessened.

The sulphur content of the furnace gases at the walls is also being determined at various locations of each furnace investigated. It adds to the general information, but there are no data at present on the relation of SO_2 or SO_3 in gases to the slagging or other failure of refractories.

FURNACE-GAS VELOCITIES

The velocity of the gases in the furnace determines first the carrying power of the gases for solids; second, the amount carried in unit time; and, third, the tendency to sweep the molten combination away to expose fresh surface for attack.

The velocities at the furnace wall in chain-grate-stoker furnaces reached 40 ft. a second at the nose of the main arch. When it is considered that this is equivalent to about 27 miles an hour, it is seen that the gases possess a considerable carrying power for finely divided solids.

The velocities in underfeed-stoker-fired furnaces are much more irregular on account of the changing condition of the fuel bed. The mean velocities over any one refractory wall are lower than in the chain-grate furnace owing to the fact that the gases are not confined under an arch, but local high velocities do occur because of cracks in the fuel bed. Values of 45 ft. a second have been measured.

The entrance velocities of the coal and air into the powdered-coal furnace as described above are very high, but the velocity is quickly dissipated in the relatively large furnace volume. The maximum velocity measured at the side wall of this furnace was about 20 ft. a second.

TABLE 1 PERCENTAGE COMPOSITION OF COAL ASHES

Constituents	Pocahontas coal	Pittsburgh coal 1	Pittsburgh coal 2	Illinois coal 1	Illinois coal 2
SiO_2	53.2	55.9	45.6	54.2	41.5
Al_2O_3	27.9	29.7	22.1	25.6	13.4
Fe_2O_3	9.7	8.3	26.3	10.1	14.3
CaO	2.9	2.0	1.7	3.2	14.6
MgO	1.8	0.6	0.8	1.1	0.9
Na_2O + K_2O (by difference)		2.3	2.2	2.8	2.9
Sulphur as SO_3		1.2	1.3	3.0	12.4
Ash in coal	6.0	9.3	10.7	13.6	16.3
Softening temp., deg. fahr.	2345	2650	2100	2390	2040

SLAG COMPOSITION

One of the most frequently considered and obviously important factors governing slag erosion is the composition of the coal ash fired to the furnace. Table 1 gives the composition of the ashes of a Pocahontas coal, two Pittsburgh coals, and two Illinois coals, burned in furnaces studied in this investigation. These are not considered representative of the fields from which they come, but are given to illustrate the variations in coal-ash composition.[5]

The Pocahontas coal has a low Fe_2O_3 content as well as of other fluxes, and, though its softening temperature is not exceptionally high, would not be expected to present serious difficulty from slagging. The Pittsburgh coal No. 1 with its low Fe_2O_3 content and high softening temperature is an ash which would also be expected to give little slag erosion, while No. 2 with its high Fe_2O_3 content and high softening temperature promises considerable difficulty from slagging.

Fig. 10 shows the interior of an underfeed-stoker-fired furnace after a period of burning the No. 1 coal. The heavy accumulation of slag on the walls will be noted. Fig. 11 shows the same furnace after a like period of burning the No. 2 coal. The walls were cleaned of slag; only a thin coating remained on the bricks except

[5] See also Fusibility of Coal Ash as Related to Clinker Formation, by W. A. Selvig, P. Nicholls, W. L. Gardner, W. E. Muntz, and E. B. Ricketts. Bull. No. 29, Carnegie Inst. of Tech., 1927, pp. 5, 7, 16, 17, 23, 51.

TABLE 2 COMPOSITION OF COAL-ASH AND SLAG SAMPLES

Item	Coal-burning equipment	Description of sample	Composition, per cent SiO_2	Al_2O_3[1]	Fe_2O_3	FeO	CaO	MgO	K_2O	Na_2O	SO_3	SiO_2 / Al_2O_3 ratio
	Forced-draft chain grates	Illinois coal ash	46.5	20.4	19.8	...	5.1	0.6	3.6[2]		4.0	2.28
		Illinois coal ash	43.5	18.1	18.0	...	8.1	0.5	3.2[2]		8.6	2.40
1		Slag from boiler tubes: material next to tubes	33.2	16.5	23.7	1.7	5.6	0.7	4.6	3.9	6.6	2.00
		Slag from boiler tubes: material on outside of pieces	40.5	17.3	22.7	7.3	6.8	1.0	1.9	1.5	0.2	2.39
	Forced-draft chain grates	Pittsburgh coal ash	42.3	20.7	29.2	...	2.8	0.7	2.8[2]		1.5	2.04
		Slags from furnace gases	33.7	18.5	34.9	...	3.5	0.7	0.6[2]		3.5	1.82
2		Slag from boiler tubes	29.6	17.1	46.2	2.9	1.8	0.4	1.8[2]		0.2	1.73
		Ash and refuse from ashpit	46.2	21.6	27.6	...	2.5	0.6	1.1[2]		0.4	2.10
	Underfeed stoker	Pittsburgh coal ash	55.9	29.7	8.3	...	2.0	0.6	2.3[2]		1.2	1.88
3		Slags from furnace gases[3]	46.2	30.6	10.1	...	1.8	0.1	...		0.6	1.51
		Ash and refuse from ashpit	57.9	30.5	7.9	...	1.3	0.5	1.8[2]		0.1	1.90
		Pittsburgh coal ash	45.6	22.1	26.3	...	1.7	0.8	2.2[2]		1.3	2.06
		Slags from furnace gases[3]	39.4	21.3	32.6	...	1.3	0.6	...		0.1	1.85
	Underfeed stoker	Ash and refuse from ashpit	48.0	23.3	23.3	...	1.8	1.0	2.0[2]		0.6	2.06
4		Slag from boiler tubes, inner portion	...	...	...	...	...	...	3.9	2.0	...	
		Slag from boiler tubes, outer portion	40.3	22.6	33.5	...	1.9	0.5	1.0[2]		0.2	1.78

[1] Al_2O_3 contains any TiO_2. [2] Alkalis, by difference. [3] Approximate composition, insufficient sample for accurate analysis.

at the surface of the fuel bed. Considerable erosion had taken place.

The No. 1 Illinois coal ash is really excellent for a coal from that field. It compares favorably with the Pocahontas coal ash. The ash of the No. 2 Illinois coal is extremely interesting owing to the exceptionally high lime content. A furnace burning coal of this type in powdered form is now being investigated. This coal is being burned with apparent success and the station reports

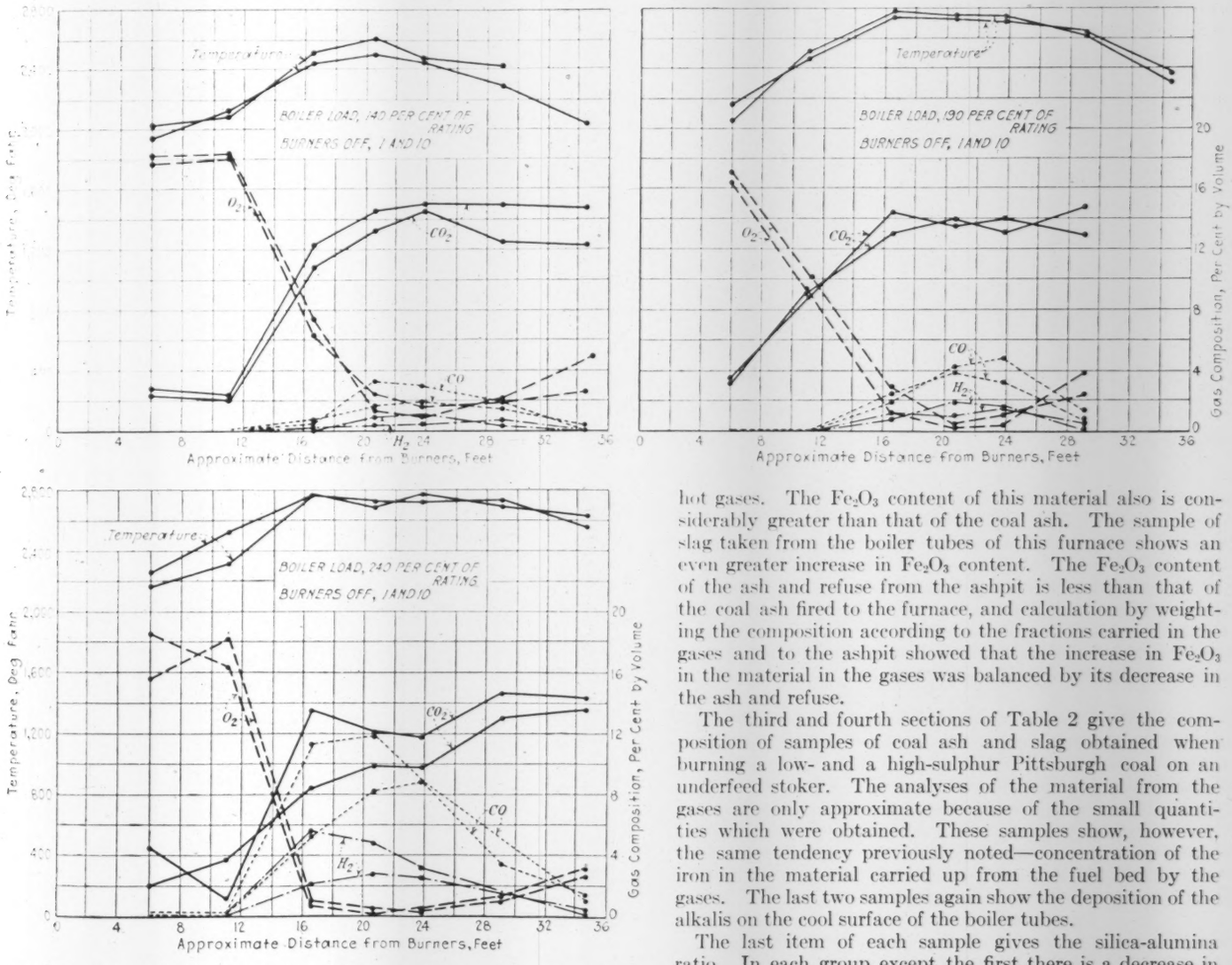

FIG. 7 FURNACE-GAS TEMPERATURE AT 6 IN. AND GAS COMPOSITION AT 4 IN. FROM SIDE WALL. POWDERED-COAL FURNACE, ILLINOIS COAL

satisfactory refractory maintenance costs under operating conditions of high boiler ratings.

COMPOSITION OF COAL ASH AND SLAG

One of the early findings in this investigation was that the composition of the coal ash fired to the furnace and that of the slag carried in the gases to be deposited on the walls differed. Samples of slag were taken from the first row of boiler tubes over a forced-draft chain-grate stoker burning Illinois coal. The composition of two samples of the ash of this coal and the composition of the material next to the boiler tubes and of the outer portion are given in the first part of Table 2. The inner material was light in color and unfused, while the outer portion was dark and glassy. Whereas the coal ash contained approximately 19 per cent of Fe_2O_3, the slags contained more than 23 per cent. The material next to the tubes contained a total of 8.5 per cent alkalis, while that on the outside contained only 3.4 per cent. The SO_3 content was also considerably higher next to the comparatively cool surface of the tubes.

The second section of Table 2 contains the analyses of samples of coal ash and slags from a furnace burning Pittsburgh coal on a forced-draft chain-grate stoker. This coal ash is notable for its high Fe_2O_3 content. The samples of slags from the furnace gases were taken into a water-cooled sampler directly from the gases and were thus quickly cooled, and should represent the composition of the material in the gases more accurately than does the material on the boiler tubes which remains in contact with the hot gases. The Fe_2O_3 content of this material also is considerably greater than that of the coal ash. The sample of slag taken from the boiler tubes of this furnace shows an even greater increase in Fe_2O_3 content. The Fe_2O_3 content of the ash and refuse from the ashpit is less than that of the coal ash fired to the furnace, and calculation by weighting the composition according to the fractions carried in the gases and to the ashpit showed that the increase in Fe_2O_3 in the material in the gases was balanced by its decrease in the ash and refuse.

The third and fourth sections of Table 2 give the composition of samples of coal ash and slag obtained when burning a low- and a high-sulphur Pittsburgh coal on an underfeed stoker. The analyses of the material from the gases are only approximate because of the small quantities which were obtained. These samples show, however, the same tendency previously noted—concentration of the iron in the material carried up from the fuel bed by the gases. The last two samples again show the deposition of the alkalis on the cool surface of the boiler tubes.

The last item of each sample gives the silica-alumina ratio. In each group except the first there is a decrease in this value for the material taken from the gases from its value for the coal ash, while it remains constant for the ash and refuse. The exact cause and significance of this change remain to be determined by the ceramic-laboratory investigators.

No analyses have been received at this time of the material taken from the gases of the powdered-coal furnace investigated. While it appears improbable that there could be a change in the composition when all of the ash is carried in the gases as it is in these furnaces, no prediction can be safely made.

SLAG QUANTITIES

A frequent and entirely logical method of designing a slag test for refractories is that of introducing powdered ash or slag into a flame to be sprayed on the bricks under test. Investigators have lacked any information on the quantities of material which are carried per unit volume of gas under service conditions, and have had to choose an arbitrary value.

An attempt has been made in this investigation to supply this information. A method of sampling has been developed wherein the gases with their accompanying solids are drawn into a water-cooled tube placed in the furnace with the open end facing the

direction of gas flow. The rate of flow is set so that the linear velocity of entrance of the gases into the sampler is approximately that of the gases at the point. A part of the solids is deposited in the tube and the remainder is collected in a cyclone-type separator and filter on the outer end. The volume of gases taken through the sampler is measured, and the weight of solids per unit volume is thus obtained.

As the volume of gases sampled was necessarily low, and as the velocity of the gases at a given point does not remain constant from minute to minute, the values given can be considered as only approximate, but it is believed, however, that they are a fair approximation.

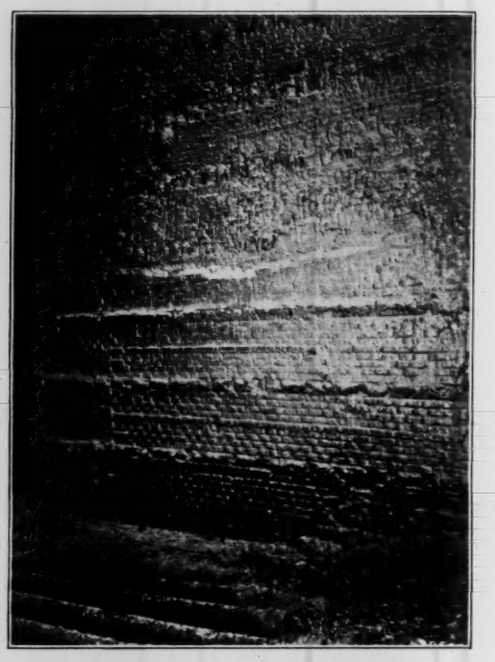

FIG. 8 SIDE WALL OF POWDERED-COAL FURNACE SHOWING EROSION

FIG. 9 SIDE WALL OF POWDERED-COAL FURNACE SHOWING AREA OF MAXIMUM EROSION

The weight of ash per 100 cu. ft. of gas in the chain-grate-stoker-fired furnace burning Pittsburgh coal at a point next to the wall and half-way between the grate and the nose of the arch, was 2 grams at 150 per cent of boiler rating. The amount in the gases at 18 to 24 in. from the wall at the same position varied from 3 grams at 130 per cent of rating to 6 grams per 100 cu. ft. at 190 per cent of rating.

The quantity of ash or slag in the gases of an underfeed-stoker-fired furnace varied from 0.2 gram at 133 per cent of rating to 1.3 grams per 100 cu. ft. at 180 per cent of rating when burning a low-sulphur Pittsburgh coal, and from 0.15 gram at 125 per cent of rating to 0.7 gram per 100 cu. ft. at 150 per cent of rating when burning a high-sulphur coal. The average values at boiler ratings of from 133 to 183 per cent for the low-sulphur coal and from 125 to 192 per cent for the high-sulphur coal were 0.8 and 0.5 gram per 100 cu. ft., respectively.

The values obtained in the powdered-coal furnace burning Illinois coal at the zone of greatest erosion were from 19 to 22 grams at 190 per cent of rating and 34 to 63 grams per 100 cu. ft. at 240 per cent of boiler rating.

Arranged in the order of increasing severity of slag erosion, the furnaces studied would be as follows: (1) underfeed stokers; (2) chain-grate stokers; (3) powdered coal. Although other reasons for this varying severity of service have been pointed out, it is interesting that the values for the quantities of slag carried in the gases fall in the same order. Of course the severity of service does not increase in the same proportion, because only a certain amount of slag can attack the walls. Beyond the point where slag is deposited as fast as the molten combination runs away, further additions probably do not increase the action on refractories.

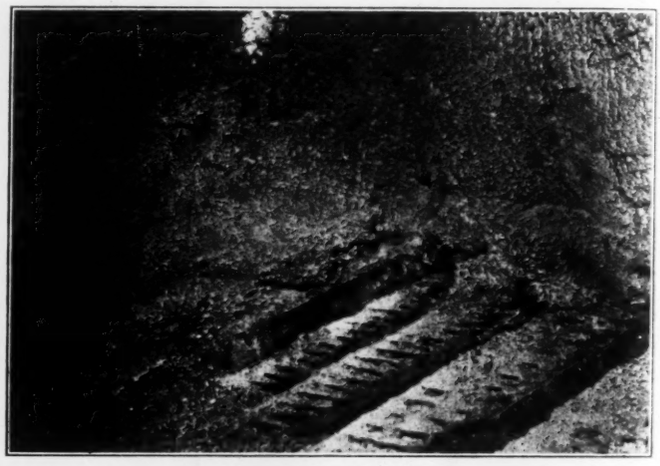

FIG. 10 INTERIOR OF UNDERFEED-STOKER-FIRED FURNACE AFTER BURNING LOW-SULPHUR PITTSBURGH COAL

FIG. 11 INTERIOR OF UNDERFEED-STOKER-FIRED FURNACE AFTER BURNING HIGH-SULPHUR PITTSBURGH COAL

CONCLUSION

The significance of six service factors—namely, refractories temperatures, furnace-gas temperatures, composition, and velocities, and slag composition and quantities—which govern the slagging of boiler-furnace refractories has been discussed and the range of their values under various conditions has been presented. Those who desire more complete data on these factors are referred to the progress reports on the investigation which are mentioned in footnote (6) on the following page.

This investigation is being carried on under the general direction of the Special Research Committee on Boiler-Furnace Refractories of the A.S.M.E., of which C. F. Hirshfeld is chairman, and of P. Nicholls, supervising fuel engineer, Bureau of Mines. Acknowledgment is made of the coöperation and helpful suggestions given by G. A. Bole, formerly superintendent of the Ceramic Experiment Station of the Bureau of Mines, and by S. J. McDowell, superintendent of the Columbus Branch of the Bureau of Standards. W. E. Rice and L. B. Berger of the Pittsburgh Experiment Station of the Bureau of Mines have been engaged on part of the field work. The analyses of coal-ash and slag samples were made in the analytical laboratory of the Bureau of Mines under the direction of W. A. Selvig, and the softening temperatures of the coal ash under the direction of H. M. Cooper.

The By-Product Processing of Coal

Discussions Dealing with Fundamental Technology of Gaseous Fuel Supply, Principles Involved in High- and Low-Temperature Coal-Carbonization Processes, Possibilities of Processes Available, Etc.

THREE papers on this subject were presented at one of the sessions of the A.S.M.E. Annual Meeting at New York last December, H. W. Brooks[1] presiding. The first, by A. C. Fieldner, dealt with the fundamental principles involved and briefly discussed the present status of representative types of processes. The second, by Wm. H. Blauvelt, discussed the principles involved in high- and low-temperature processes of carbonizing coal, the difference in the products obtained therefrom, and the fields for each process. The third, by R. S. McBride, discussed the fundamental technology and economics of gaseous-fuel supply, dealing with the present status and possibilities of systems available for making gas for use in mechanical and chemical-process industries. These papers were published in full in the Mid-November, 1926, issue of MECHANICAL ENGINEERING, pages 1210–1227. An abstract of the discussion which they drew forth immediately follows.

THE LOW-TEMPERATURE CARBONIZATION OF COALS

Discussing Mr. Fieldner's paper on the above subject, F. C. Weber[2] thought that the availability of large supplies of petroleum for some considerable time in America seemed to obviate the necessity or desirability of producing oils through low-temperature carbonization of bituminous coals. The time would undoubtedly come, however, when this would be necessary, and it was therefore desirable to have the problem in mind in a general way. The process, however, might serve, in the near future, as a medium for the development of certain low-grade non-coking coals into a good, usable, and approximately smokeless domestic fuel. It might also prove to be sound economics to treat certain coals in connection with power-plant operation.

In the paper presented by Mr. McBride it was stated that if a gas of something in excess of 300 B.t.u. were to be considered for general city use, the distribution cost usually would preclude its economic supply. Mr. Weber questioned whether Mr. McBride had any actual proof of this statement. He had found in actual practice that substantial reductions in the B.t.u. capacity of city gas had very little effect on the performance of the distribution system. Furthermore he knew nothing to indicate that British gas undertakings in going to very low B.t.u. standards, in conformity to the Therm Regulation Act, had encountered appreciable distribution-system troubles. Very few British companies supplied gas as high as 500 B.t.u., and several used values as low as 275. Surely if the British distribution systems had required extensive and expensive alterations to accommodate themselves to the low B.t.u. value, they would not have gone to these low standards.

A. G. Christie[3] wrote that while one of the objectives of this form of coal distillation that was common to all processes was the production of liquid fuel as a by-product, and while such fuel had a peculiar significance and value in certain European countries, Americans should have a definite appreciation of the fact that at present and for some time to come, liquid fuels must be secondary by-products in such processes. This was obvious from the fact that our supplies of petroleum would not be exhausted for many years since new and improved methods would greatly increase gasoline yields from the crude oil, and also since further supplies of petroleum would undoubtedly be discovered. Even if our petroleum supplies were diminished there were the huge supplies of oil shales in the western states that could be treated to yield products at competing prices with the liquid fuels from low-temperature carbonization.

Most processes in America aimed to produce smokeless fuel for domestic consumers or a semi-coke which might serve as boiler fuel. Smokeless fuel as a primary product had its limitations. It had to sell to the consumer at a higher price than bituminous coal on account of the costs of manufacture and at a less cost than anthracite, for the consumer would not use coke if hard coal could be bought at the same cost. The heating value of both would be substantially the same. It was very difficult in cities where soft coal was burned almost exclusively for domestic purposes, to induce the buyer to pay higher prices for a smokeless fuel than he now paid for bituminous. This led to the conclusion that the market for smokeless fuel of this character would be in those parts of the country where people had been accustomed to burning anthracite, and where on account of the rising prices of such fuel, a cheaper substitute was desired. Such consumers would buy the semi-coke and would take the necessary trouble to fire it, in order to secure the benefits of freedom from smoke and dirt. The person who had always lived in a town where soft coal had been burned, did not place such a high value on cleanliness and a smokeless atmosphere. Hence the market for such smokeless fuels was, at least for the present, confined to districts now using anthracite. The operator of a modern central station bought heat value in his fuel. Hence he could not afford to pay more for his semi-coke than for the equivalent in raw coal. Smokelessness in the fuel had no definite value in such a plant. When low-temperature-carbonization plants were installed to provide treated fuel for central stations, the liquid and gaseous by-products must have values that would pay operating and fixed charges, and profit on the investment in the distillation plant. This situation exists at a few places, as for instance at Lakeside Station, Milwaukee, and might warrant this treatment of coal at still other central stations or gas plants. There was another economic side to this problem. The liquid and gaseous by-products naturally varied in quantity with the percentage of volatile matter in the coal. The values of such products per gallon or per cubic foot, in general, did not vary widely over this country. It would therefore be evident that of similar high-volatile coals, the lower-priced ones would show the greater economic yields from low-temperature distillation. This treatment would therefore seem more applicable to cheap washed slack coal or to the cheap bituminous and lignite coals of the Middle West and the South.

J. A. Perry[4] wrote that in this country, as in Europe, there were available considerable quantities of slack and other low-grade coals and refuse, which could be bought cheaply at the mines.

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[5] Refractories Service Conditions in Furnaces Burning Pittsburgh Coal on Chain Grates, by R. A. Sherman and W. E. Rice. MECHANICAL ENGINEERING, vol. 48, pp. 1115–1122, Nov., 1926.

An Investigation of Boiler-Furnace Conditions as Related to Refractories Service, by R. A. Sherman, W. E. Rice, and L. B. Berger. MECHANICAL ENGINEERING, vol. 48, pp. 1389–1396, Dec., 1926.

Temperatures in Powdered Coal Furnaces Equipped with Extended Radiant-Heat-Absorbing Surface, by R. A. Sherman. MECHANICAL ENGINEERING, vol. 49, pp. 335–338, April, 1927.

[1] Consulting Engineer, Erie City Iron Works, Erie, Pa. Mem. A.S.M.E.

[2] New York, N. Y.

[3] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

[4] United Gas Improvement Co., Philadelphia, Pa.

Bearing in mind the freight to destination, the ash content, the moisture content, and other drawbacks, such coals did not promise much in the way of economy for gas making, even at low prices, compared with paying the much higher prices that prevailed for the better low-ash and low-moisture-content coals that were now generally used for coking and gas-making purposes, although it was the dream of the inventor to find processes which would make the low grades profitable.

He had seen in Germany, near Essen, one of the most highly developed inclined rotary types of low-temperature-carbonization plants, and was told by the builders and operators that the investment cost per ton of coal handled per day was practically the same as for the same capacity of high-temperature coke ovens. The coke produced was too soft to withstand the necessary handling and transportation or to make it satisfactory for use in this country as a domestic fuel. As regarded the tars or oils produced, it would appear that in this country these tars, if once produced on any large or commercial scale, would certainly at first be mostly used as fuel, and that for some time to come they were not likely to have a value much greater than 4 cents per gallon, at the place of production. When it came to refining the tar, and then cracking up, at the most perhaps one-third of the total amount of tar produced, to make more valuable oils, then these separating and cracking costs, as well as investment costs, had to be taken into consideration in estimating the value of the entire tar production. In England he had learned that one of the large coke-oven operators had shown that the cheap coals available in England for low-temperature processes could be easily and cheaply blended with other coals, and carbonized in existing types of ovens to produce a good, hard, and dense coke.

Examining the gas production by low-temperature processes we should consider whether we were really getting value received for the so-called greatly increased yield of tars and oils. With 2000 lb. of good gas coal, in a high-temperature carbonization plant we could produce about 11,500 ft. of 560-B.t.u. gas, and 12 gal. of tar. If the same 2000 lb. of coal, in a perfected Barnsley type of low-temperature plant, could produce 5620 ft. of 700-B.t.u. gas, and over 20 gal. of crude tars and oils, as well as make a coke that could be used for blue-gas making or for sale, there was still a question as to the relative value of these two processes for making gas, if gas oil were not available at reasonable prices for enriching purposes. If in each case we made blue gas from the coke produced, to mix with the coal gas, and obtained in each case a 400-B.t.u. mixed gas, we should have:

By the High-Temperature Process		
Coal gas from carbonization of coal, 560-B.t.u. gas	11,500 cu. ft.	
Blue gas made from own coke, requiring 33 lb. coke per M ft. of 300-B.t.u. gas	18,400 cu. ft.	
Total 400-B.t.u. gas, no oil enrichment		29,900 cu. ft.
By the Low-Temperature Process		
Coal gas from low-temperature plant, 700-B.t.u. gas	5,620 cu. ft.	
Blue gas made from own coke, 300-B.t.u. gas	16,840 cu. ft.	
Lightly carbureted water gas, to make up same quantity of total 400-B.t.u. gas as is the case in high-temperature plant, requiring 1.25 gal. of oil and 36 lb. of coke per M ft., 400-B.t.u. gas	7,440 cu. ft.	
Total gas, including 7440 cu. ft. of carbureted water gas		29,900 cu. ft.
Equivalent gas oil saved by high-temperature process per ton of coal carbonized (7.44 × 1.25)		9.30 gal.
Extra coke required for blue gas and carbureted water gas per ton of coal carbonized (low-temperature process)		189 lb.

These 9.30 gal. of equivalent gas-oil enrichment, and 189 lb. of coke saved, were certainly of more value to the gas man than (20 — 12 =) 8 gal. of additional tar, and the saving of perhaps 100 lb. less producer fuel for carbonization in the low-temperature plant.

In fact, if the total production of low-temperature tars ran as high as 30 gal. per ton of coal treated, and if after distilling into suitable fractions there were a recovery of only 33⅓ per cent, or 10 gal. of oil suitable for cracking purposes, then to the gas man, other things being equal, the high-temperature process would still obtain the most favorable consideration. As to the beneficiating of cheap coals, and especially for powdered-fuel firing, drying and preheating powdered coal and then semi-carbonizing with hot recirculated gases, it would seem that, if this scheme could be worked out mechanically and otherwise satisfactorily, all such surplus gas could be absorbed at a fair price by the gas utility of the district.

G. I. Vincent[5] drew from Mr. Fieldner's paper the apparent impression that high-temperature plants did not make a domestic fuel which would compare favorably with anthracite coal. This was far from being a fact. His company, with the keenest anthracite competition, had built up a coke business for domestic use in Syracuse to about 90,000 tons per annum.

Mr. Fieldner, in his closure, noted that Professor Christie had made the assumption that the natural market for a low-temperature coke was in the anthracite field, that was, on the Eastern Seaboard where people were accustomed to burning anthracite. He believed that a process of education had been taking place in the Middle West through the use of high-temperature coke. The production of high-temperature coke often had made considerable progress as a domestic fuel. The material had been better sized than before, and the manufacturers so selected their coals as to produce a material that lent itself well to furnace use. Therefore people had become accustomed to smokeless fuel, and many of them had burned oil before. It was not quite as difficult now to introduce a higher-priced fuel in the bituminous field as it had been in the past.

THE DISTILLATION OF COAL

K. M. Simpson,[6] thought that Mr. Blauvelt had rightly placed emphasis on the use of low-temperature char as boiler fuel, especially in connection with pulverized-coal firing. In this method of firing it was either necessary or desirable to dry coal before pulverizing, and it was only a short step from drying to low-temperature distillation. Coal that had been subjected to low-temperature treatment was absolutely dry; that was to say, not only had the surface moisture been removed, but also the so-called water of constitution. All of the surface water could be removed by continued heating at 105 deg. cent., and the water of constitution was removed by heating to higher temperatures. Water of constitution began to form at approximately 200 deg. cent. and continued to form and to be evolved up to 600 deg. cent. Violent evolution of water of constitution took place at 300 deg. cent. The use of thoroughly dry coal in pulverized firing was highly desirable, not only because moisture was a diluent, but also because its action was actively harmful. The action of water could best be illustrated by considering the high-moisture coals of the Middle West. When these coals were burned under boilers, the sulphur united with the moisture in the coal, with the result that a sooty deposit contained a high percentage of acid condensed on the tubes of the economizers. This action was responsible for the corrosion of the economizer tubes, which was one of the serious troubles that arose in connection with burning these high-moisture coals. It had been customary to consider that the corrosion of iron and steel during the combustion of coal was due to the direct action of the sulphur, but it was probable that there was very little deleterious action due the combustion of sulphur in the absence of moisture.

J. A. Perry[7] wrote that great progress had been made during the past thirty years in distilling or carbonizing coal on a large scale in modern high-temperature by-product ovens. True, the aim had been principally to perfect the manufacture of a high grade of coke for metallurgical purposes, but along with this coke perfection had come the ability to blend and use cheaper grades of coal, to produce high yields and good grades of gas, tar, and ammonia. Nor had the development and market of coke suitable for house heating and industrial uses been found difficult. The high-temperature processes stood out as reliable, well developed, and now at our hand ready to use on a commercial basis. It would appear, however, that the low-temperature processes, of perhaps two or more types, promised during the next five to ten years to be developed satisfactorily in a mechanical way. For the permanent commercial success of any low-temperature process, development, and perfection

[5] Vice-Pres., Syracuse Lighting Company, Syracuse, N. Y. Mem. A.S.M.E.

[6] New York, N. Y.

[7] United Gas Improvement Co., Philadelphia, Pa.

of suitable drying and preheating apparatus was necessary. After suitable drying and preheating came the semi-coking apparatus, after which it appeared that we had a fuel that would be used principally as a smokeless or powdered boiler fuel. By briquetting the semi-coke and recarbonizing, making four processes in all, McIntyre showed that he had produced a hard and dense coke, which would compete with anthracite coal and high-temperature coke for use as a domestic fuel. With the perfection of suitable drying and preheating apparatus we could expect by its use to increase the throughput on high-temperature plants from 25 to 50 per cent. The gases and vapors should be driven off faster, tar yield would be greater, and the gas perhaps of higher grade. Again, if throughput was the big factor, using a cheap grade of coal to produce a coke principally for boiler-fuel use, then it was the writer's opinion that the high-temperature continuous retort and oven plants would be designed and operated at high rates for this purpose, using the cheap coals available, slightly blended with other coals. Looking at the gas and coke side of the question from the gas point of view, first, 11,500 cu. ft. of 560-B.t.u. gas by the high-temperature process was worth approximately 10 gallons of equivalent gas oil more than 5620 ft. of 700-B.t.u. low-temperature gas. Undoubtedly the high-temperature processes cracked up into permanent gases more of the tars and oils, and apparently at a high efficiency. The writer asked whether it was probable that as much as 10 gallons of equivalent gas oil would be obtained, by the refining process, out of a yield of 20 to 30 gallons of tar from some low-temperature process. Again, for making a uniform grade of blue gas, of about 300 B.t.u. there was no better fuel than good, hard, dense coke, that could be stored and handled satisfactorily.

G. E. Rohmer[8] spoke of experiments he had viewed in Germany, and of a large plant for low-temperature distillation now being put up in the South.

G. A. Orrok[9] said he had noted in the slides shown that in high-temperature distillation 25 lb. per ton of ammonium sulphate and in low-temperature distillation around 12 lb. per ton had been obtained. In ordinary coals there should be enough nitrogen to give from 70 to 80 lb. of ammonium sulphate, and perhaps more. But, as a matter of fact, in the high-temperature work a lot of cyanide was made and usually wasted. More than half of the nitrogen that was in the coal went into the cyanide. But the price of ammonium sulphate, about $50 per ton, was not high enough to really pay for saving it. Nor was the cyanide worth saving. Also there was no profit to the manufacturer in selling tar to the tar distiller at about five cents a pound while it was worth as fuel from seven to seven and one-half cents. Raw coal burnt on a stoker or in powdered form in the power station could give an efficiency of 88 per cent over a year.

The hydrogen loss, the loss in the latent heat of water made by burning the hydrogen, and the loss by moisture that was in the coal might amount to five per cent, so we could get nearly 95 per cent of the available heat units in the coal in the making of power. High-temperature coke worked very well, but did not give the efficiency of raw coal. Possibly with the powdered form we might be able to secure equal efficiency in the future.

Mr. Blauvelt, in his closure, agreed with discussers that the economic aspect of the processes was, after all, the final measure. We could not expect any process in which fine-grade screened lump bituminous coal was used to give results comparable with processes where screened bituminous coal was utilized in its raw state. In the Illinois fields, in the lignite fields of Texas or the Dakotas, there was a large amount of material which was unsalable in its raw state, and could therefore be purchased very cheaply. Such material, which could be made to produce a valuable fuel by processing, should be particularly interesting for low-temperature distillation processes. The difference between costs and receipts was not very great. The cost of the apparatus had to be carefully considered. There were several low-temperature processes now being developed where the price of apparatus was low. It was in this direction that success lay. In regard to Professor Christie's remarks about oil shales, there was an important distinction between the processes of making oil from shale, and the processes for making oil from coal. In the former case there was practically the same bulk residue as in the latter. But in the processing of the coal a valuable by-product was obtained instead of a worthless one. For this reason the securing of oil from coal was a little more promising than the securing of oil from shale. Mr. Orrok had pointed out that the hydrogen content of the coal was responsible for the loss of four or five per cent efficiency. He wished to point out to Mr. Orrok that the low-temperature carbonization process offered an opportunity of saving this four or five per cent, because it allowed the hydrocarbons to be taken out of the coal and turned into valuable oil. He did not think that the low-temperature distillation process would exclude all other processes, but it would certainly take its proper place in the arts.

[8] Consulting Metallurgist, New York, N. Y.

[9] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

COMPLETE GASIFICATION OF BITUMINOUS COAL

Discussing Mr. McBride's paper of the above title, J. A. Perry[10] wrote that the present methods of gasification of bituminous coal were well adapted for industrial purposes and highly efficient. He thought that coöperation with a public-utility company was the most likely method of securing cheap gas for industrial purposes. A large industrial consumer could arrange with a public-utility company for a supply of gas below city standards, but entirely suitable for his needs. He thought the use of the best grades of bituminous coal would soon pass, and that where good lump coke was available at a cost not exceeding the cost of bituminous coal by more than $1.00 a ton, the use of the coke was preferable for the manufacture of either blue gas or carbureted water gas, because its use would mean less plant investment and carrying charges. As time went on and more large coke-oven installations were put into use, he thought that coke would certainly be produced at a cost not exceeding that of the bituminous coal by more than $1.00 a ton. He disagreed with Mr. McBride's statement that complete gasification of all the coke would produce a mixture of oven gas and blue water gas rather too lean to permit of economic city supply. A 400-B.t.u. standard of gas supply would permit, in his opinion, of a satisfactory supply of this kind of gas to ordinary customers, and at the same time it would permit the use of the same kind of gas through the same mains by industrial customers. Distribution or delivery of this 400-B.t.u. gas would only mean a very reasonable expenditure for additional and larger trunk mains, pushers, and governor stations. So far as flexibility and operating possibilities for various load conditions were concerned, Mr. Perry gave the following illustrations:

Take, first, a coke-oven plant of, say, 1000 tons capacity per day, and designed to carbonize either with coke producer gas or with the 560-B.t.u. oven gas:

Capacity to make 560-B.t.u. gas per day:	
Coke-producer fired	11,500,000 cu. ft.
Oven-gas fired	7,000,000 cu. ft.
A—Maximum Day Capacity for 400-B.t.u. Gas	32,000,000 cu. ft.
Operating ovens to full capacity of 4000 tons of coal per day	
Operating approximately 36 per cent 560-B.t.u. coke-oven gas for supply	
Operating approximately 64 per cent 300-B.t.u. coke blue gas for supply	
Operating with coke-fired producers for heating carbonizing plant, with 130-B.t.u. producer gas	
B—Minimum Day Capacity 400-B.t.u. Gas	19,400,000 cu. ft
Operating ovens to full capacity to 1000 tons coal per day	
Operating 36 per cent 560-B.t.u. oven gas for supply	
Operating 64 per cent 300-B.t.u. coke blue gas for supply	
Operating carbonizing plant to full capacity, with 560-B.t.u. oven gas	
C—Further reduction of minimum day capacity for three months, say, operating ovens to two-thirds of capacity of B, to produce 400-B.t.u. gas—say	13,000,000 cu. ft.
D—Further maximum daily capacity of plant making 33⅓ per cent of output 400-B.t.u. carbureted water gas for heavy-send-out season—add 50 per cent to A, or	48,000,000 cu. ft.
E—Further variations can be made of economical operating capacities by slacking down operation A, or by using 400-B.t.u. gas for heating the ovens.	
F—Ratio of maximum day send-out to minimum day send-out, no carbureted water gas, = A ÷ C =	2.46
G—Ratio of maximum day send-out to minimum day send-out using carbureted water gas for peak loads, = D ÷ C =	3.70

[10] United Gas Improvement Co., Philadelphia, Pa.

Adding 50 per cent of 400-B.t.u. carbureted water gas to a mixture of 400-B.t.u. coke-oven and blue gas would only increase the gravity from approximately 0.50 to approximately 0.53, so we should have no difficulties from this small change.

Excess Coke Produced Per Day:

Operating as A	approximately 170 tons
Operating as B	approximately 480 tons
Operating as C	approximately 320 tons
Operating as D (requires from storage)	100 tons.

It would appear from this that a little variation or balancing up of the operating conditions would make it possible to use up all the coke produced; or that any reasonable surplus could be sold, or used up in making blue gas to furnish a special supply of 300-B.t.u. gas to one or more large industrial customers.

S. E. Friedeberg[11] wrote that it was apparently cheaper for the individual plant to install its own gas plant and use hot producer gas when possible, than to purchase from the public-utility companies. The steel, glass, and ceramic industries in this country were today the largest users of producer gas, and most of them used it raw (that was, without cleaning it, and at about 1200 deg. fahr.) in their industrial furnaces and kilns. The above industries gasified somewhere around 25,000,000 tons of coal a year, which would correspond to about 400,000 billion B.t.u. a year. The gas companies of this country, however, from the Gas Association's figures, only consumed in 1925, 12,200,000 tons of coal and coke. This produced 300 billion cu. ft. of city gas, which corresponded to 160,000 billion B.t.u.

The manufacture of raw producer gas was much cheaper than the purchase of city gas. The estimate of fuel costs on a basis of cents per million B.t.u. should be standard. The cheapness of producer gas as compared with city gas could be demonstrated by the following example. A standard make of a fully automatic gas producer 8 ft. in diameter, would cost, completely installed with necessary coal-handling equipment, platforms, etc., about $12,000. This equipment would easily gasify completely 1000 lb. of coal per hour at 80 per cent efficiency, which would amount to 85 billion B.t.u. per year. The cost would be

Fuel, 12 tons coal per day at $6 ton fired, 350 days	$25,200
Labor	4,200
Depreciation 20 per cent, allowing 5 years to pay for equipment	2,400
Interest on investment	720
Maintenance, electricity, and water	480
The cost for 85 billion B.t.u. would then be	$33,000

which amounted to 39 cents per million B.t.u. After five years this cost would drop to 31 cents per million B.t.u. City gas of 525 B.t.u. at 60 cents per 1000 cu. ft. would amount to $1.15 per million B.t.u. This price was about 300 per cent greater than the initial cost of producer gas and about 375 per cent greater than the final cost. Some industrial plants had gone to the extent of installing equipment to clean the gas. This would increase the cost about 25 per cent. However, it would then have the advantages of clean city gas, except as regarded lower B.t.u. value per cu. ft. City gas due to its high B.t.u. value burned with high flame temperature and therefore very efficiently. Thus less combustion space was required in the furnaces and it was not necessary to use hot air. However, for maximum efficiency in furnaces or kilns, regenerators or recuperators should be used. Most industrial furnaces sold today had to operate on the recuperator principle, and with air at 1400 deg. fahr. Hot producer gas with hot air could be burnt almost as efficiently as city gas with hot air.

Preference might be shown for city gas because it was supplied clean. This of course eliminated soot-cleaning periods. City gas was also more readily distributed than producer gas without loss in heat value. Hot producer gas might be readily distributed through long flues by insulating the flues; by installing soot collectors, soot might be eliminated from the producer gas without loss in heat value.

Farley G. Clark[12] wrote that although greatly interested in all aspects of the processing of coal and in the allied hydrogenation of coal, his immediate concern was the adaptation of these methods to the benefit of the public utility, and in particular he desired to point out some of the benefits to be derived concurrently with the evolution of the art of processing coal by combining or coördinating the steam plant and the gas plant. Referring directly to Mr. McBride's paper, Mr. Clark stated that the author did not bring out the fact that the gas made by the oxygen blast, although a water gas, was not the 50 per cent carbon monoxide and 50 per cent hydrogen mixture of the intermittent process but was theoretically two-thirds carbon monoxide and one-third hydrogen, and therefore about 30 per cent heavier. Hydrogen was the important item if it could be had cheaply enough; which he would show to be the case. Cheap hydrogen could be used to restore the 50–50 volume of the gas made in the oxygen blast generator.

He doubted whether it would be advisable or possible under regulatory conditions to sell blue gas on a B.t.u. basis, but it should be both possible and economical to distribute a mixture of low-temperature-carbonization gas, tar-hydrogenation gas, and the 50–50 water gas, and regulate its B.t.u. value to any desired point between 400 and 450. Owing to the lightness of this gas the greater volume necessary as compared with 550-B.t.u. gas could be handled in the same distribution system without trouble or change.

Assuming that when coal cost $5 per ton the coke was worth $8 per ton, it was fair to say that blue water gas by the intermittent process would cost 30 cents per 1000 cu. ft. in the holder. With this same coke we could afford to pay over 60 cents per 1000 cu. ft. for the oxygen blast, and if we could have both oxygen and hydrogen at 35 cents per 1000 cu. ft., and the oxygen actually had an economic value of 60 cents, then the hydrogen could be used at 22.5 cents per 1000 cu. ft. although it need not be used at less than 30 cents to be an economical diluent of the 70–30 gas made in the oxygen blast generator.

The author had stated that oxygen even under the most favorable conditions was likely to cost much more than $1 per 1000 cu. ft. The reply was that when a.c. energy could be had at 3 mills per kw-hr., the gases oxygen and hydrogen made electrolytically by the continuous use of power could be had at 30 cents per 1000 cu. ft., and at 35 cents when made with the energy available one-third of the time. It was only necessary to use all of the electrolytic gas in the manufacturing gas processes to maintain this balance, for if only oxygen were used its cost would be 90 cents to $1.05 per 1000 cu. ft., and for hydrogen 45 to 52.5 cents per 1000 cu. ft.

The economical coördination of the steam plant and the gas plant would require that both be operated at some average rate that might change between one season and another but would be fairly constant from day to day. The problem would be to use the off-peak capacity of the steam plant in gas manufacture and make use of the by-products of gas manufacture in generating electrical energy. The steam plant would probably have a generating capacity three times its average loading from sales of energy, and would have a seasonal winter peak from 20 to 30 per cent greater than the summer peak. The load factor could probably be increased 30 per cent without any requirement for reserve apparatus. For a 300,000-kw. plant this would be an annual increase of about 450,000,000 kw-hr. For a modern plant the normal operating economy would be 14,000 B.t.u. per kw-hr., and the high-load-factor operation might bring the economy to 12,000 B.t.u., per kw-hr. With 13,500-B.t.u. coal at $5 per ton the off-peak kilowatt-hour bearing no capital charges would cost less than 3 mills.

With a complete installation of converters and electrolytic cells taken at 15 per cent, the cost of oxygen and hydrogen made from this off-peak power would not exceed 35 cents per 1000 cu. ft. The steam plant would be able to use 10-cent water gas as efficiently as the raw coal at $5 per ton, and could use the low-temperature coke at $6 per ton just as efficiently as the coal. The cheap hydrogen could be used as the cooling medium for the generators and produce economy, and there was the possibility that a saving in the cost of the generators of 25 per cent might ultimately result.

The gas plant would probably be called upon to deliver on a maximum day fully five times the gas demanded for same minimum day during the summer. The curve of gas demand and that of electricity demands were not very similar, although both dipped from winter to summer and rose from summer to winter. Aside from the transient demands the average winter load could hardly be more than double the summer load. Gas storage permitted changes in the rate of gas manufacture by easy stages, but the investment for this was considerable. His proposal, briefly stated,

[11] Engineer in Charge of Construction, American Shaw Gas Kiln Corp., New York, N. Y. Jun. A.S.M.E.

[12] Consulting Engineer, Toronto, Ont. Mem. A.S.M.E.

was to use the off-peak and possibly some of the seasonal power in the manufacture of gas. The transfer agent was the electrolytic cell. The power plant would use all coke produced in the gas plant not needed for gas manufacture, and the ultimate result would be a processing of all coal needed for electrical-energy generation.

The gas plant might use some sort of a low-temperature carbonization process to produce high-B.t.u. gas, primary tar, some oil, and coke. The gas would go to the holder, but the tar and the oil and possibly some of the coke would be hydrogenated, resulting in valuable oil recovery, refuse oil and tar, high-B.t.u. gas for the holder, and dry coke. Part of the coke together with steam bled from a low-pressure stage of the steam turbines and the oxygen would produce the amount of water gas needed. The mixture of gases made would not exceed the base load. The remainder of the coke would be sent to the power plant where it would be burned partly as pulverized coke or fed hot to chain-grate stokers. Cheap oxygen and hydrogen had many other uses, and opportunities for using the gases more profitably must not be lost sight of.

Martin J. Conway[13] wrote that the scheme which would lend itself most readily to steel-plant conditions was the complete gasification of bituminous or semi-bituminous coal, producing 400-B.t.u. gas with possibly the recovery of a fuel tar and sulphate of ammonia as by-products.

The overall efficiency of the average steel-plant gas producer was around 80 per cent, or in round figures, delivering 10,000 B.t.u. in the form of gas at the furnace per pound of coal gasified. There were a large number of steel plants which gasified 500 tons or more of bituminous coal per day, which could simplify their fuel distribution by producing 400-B.t.u. gas at a central plant instead of 130- to 150-B.t.u. gas in scattered gas producers. Such a scheme would give the fuel tar for isolated furnaces and the gas for larger units, such as open-hearth and soaking-pit furnaces, and would not present more of a piping problem than the present cooling water supply.

However, the increased efficiency of the central plant over the present individual gas producer would have to be great to warrant scrapping the present gas producers and mains, and the initial cost would need to be comparatively low together with a low cost of operation and maintenance.

H. K. Seeley[14] called attention to the fact that the increase in thermal efficiency of any of the proposed methods over the present methods of treating our raw materials would be relatively slight, because the thermal efficiencies of present methods were already high. Any appreciable reduction in cost would have to be made, therefore, through reduced investment and labor charges. The complete-gasification processes already in use would be more useful if the heating value required of the gas produced could be lessened. Effort should be directed to developing equipment which could use efficiently gas of from 350 to 370 B.t.u. per cu. ft. and in educating the public to the use of such gas.

Theodore B. J. Merkt,[15] stated that it was important in comparing various fuels to evaluate the advantages and disadvantages of each. The work of the Research Bureau of the American Society of Heating and Ventilating Engineers in showing the effect of various atmospheric conditions upon the human worker certainly challenged any opinion that certain intangible advantages could not be evaluated when considering such items as spoilage, inferior quality, better working conditions and cleaner plants. The formulation of a heat-processing code designed to take all heat-process costs into consideration in figuring the "Fuel Costs per Unit of Product," was immediately necessary if we were to attack this problem from the engineering viewpoint. Mr. McBride had stated that industrial users' appliances, under close supervision by skilled men, could be easily adjusted to meet seasonal changes necessitated by mixed gas systems. It was important to realize that this could be done only with the most efficient types of gas-burning appliances and heat controls, as the two determining factors were the regulation of temperature and the provision for quickly adjusting the atmosphere created by the flue gases, conditions which could not be met as easily with any other kind of fuel. Mr. Friedeberg did not take into consideration in the compilation of his fuel costs the amount of money a manufacturer could use in running his business, but which was needed for investment in a producer-gas plant on his premises. It was important that the money which could be gained by such turnover should be credited as an additional item in the cost of operation of a producer-gas plant.

Mr. McBride, in closing, said that no generalization was possible as to whether it was better to coöperate with a public-utility company or to build an individual gas plant. In response to Mr. Weber's comments, he said that if 200-B.t.u. or 300-B.t.u. gas could be delivered at a lower cost per million B.t.u., then he would not oppose the lower standard. He believed the greatest handicap in the public-utility gas industry to be the uneconomic rate schedule which still prevailed, and by which the industrial gas user paid the same flat rate per 1000 cu. ft. as the apartment dweller.

[13] Fuel Engineer, Lukens Steel Company, Coatesville, Pa. Mem. A.S.M.E.

[14] Superintendent of Gas Production, Syracuse Lighting Company, Syracuse, N. Y.

[15] Industrial Engineer, Brooklyn Union Gas Co., Brooklyn, N. Y. Mem. A.S.M.E.

A New Power Station at Carlisle, England

AT PRESENT the boiler house contains four Stirling boilers, equipped with integral superheaters and underfeed self-contained stokers, together with a Green's economizer and Davidson induced-draft fan. With the moderate-sized units which were to be used in the turbine room, it was found possible to install boiler plant of sufficient capacity on one side of a boiler house of equal length to the turbine room, leaving the other side available for windows. The steam-raising capacity of each of the boilers is 30,000 lb. per hour, while a boiler to be transferred from the James Street station has a capacity of 25,000 lb. per hr. The steam pressure is 255 lb. per sq. in., and the total steam temperature 650 deg. fahr. At its entrance to the economizers, the feedwater temperature is 140 deg. fahr. All the boilers are capable of 20 per cent overload, and all fans, stokers, conveyors, etc., are naturally proportioned accordingly. Each boiler has a heating surface of 8303 sq. ft., while the superheaters have a surface of 992 sq. ft. The economizers consist of 240 10-ft. pipes. A chimney, 70 ft. high and 6 ft. 6 in. in diameter, is shared by two boiler units. A covered annex accommodates the coal-receiving and handling plant, and also the rail siding which terminates in the turbine room, thus permitting heavy plant to be placed directly under the turbine house crane. For working this siding a fireless steam-storage locomotive is provided. A hopper below railway lines receives the coal from the railway wagons, and from this hopper the coal is fed into a gravity bucket conveyor for distribution to storage bunkers in front of the various boilers. A water-submerged conveyor removes the ashes continuously, and delivers them into an elevated ash hopper outside the boiler house. Ashes and soot can also be removed by trucks running on rails in the boiler basement.

From what has been said concerning the boiler plant, it will have been gathered that the steam conditions of this new station are not by any means exceptional, the object aimed at, we are told, being to obtain the highest thermal efficiency compatible with reasonable capital and maintenance costs. Steam is admitted to the new 6000-kw. set at a pressure of 250 lb. per sq. in., and superheated to a total temperature of 650 deg. fahr., and it exhausts into a vacuum of 29.1 in. Branches are arranged on the turbine so that steam can be bled for feed-heating purposes. The speed of the machine is 3000 r.p.m., and it is designed to develop 6500 kva.—6000 kw. at 0.8 power factor—at 3300 volts and 50 cycles per sec., the overload capacity being 25 per cent for two hours. The turbine is a Curtis ten-stage machine, the first stage having two rows and each of the remaining stages a single row of blades.

There are two condensate pumps, each driven by a 13-hp. squirrel-cage motor, running at a speed of 1440 r.p.m. Each pump is capable of dealing with the full quantity of condensate, and a control system is provided to insure a steady flow of the condensate and to prevent the pumps depriming. There are two Mirrlees steam ejector air pumps with an intermediate surface condenser between the two stages of the pumps. Each pump has a capacity of 35.5 lb. of air per hr., and the discharge from the second stage of the ejectors passes to the small surface heater, so that practically the whole of the heat from the operating steam is recovered in the condensate or feed water.—*The Engineer*, May 27, 1927, p. 573.

Development and Testing of Railway Draft Gears

Problems Encountered in Dealing with Forces Set Up by Heavy Cars During Impact as in Hump-Yard Switching and in Starting and Stopping Heavy Trains—Present-Day Draft Gears—Characteristics Required in Draft Gears—Methods of Testing Draft Gears—Test Data, Etc.

By A. F. STUEBING,[1] NEW YORK, N. Y.

THE movement of freight traffic on the railroads of this country is notable for the large loads handled in individual cars and in trains. The ordinary 50-ton car carries an actual load of about 125,000 lb., and has a total loaded weight of 169,000 lb. The largest freight cars in regular service are the Virginian Railway's special gondolas with a capacity of 240,000 lb. and a total weight of 315,000 lb. Train loadings vary greatly, but 100-car trains of 6000 to 8000 tons are not unusual on lines with moderate grades, and on the Virginian maximum train loads of 16,000 tons have been handled.

In what follows, the author will discuss some of the problems encountered in dealing with the forces set up by these large masses during impact, as in hump-yard switching, and in starting and stopping heavy trains. Most of this discussion will be devoted to the draft gear which cushions the forces between cars. This is so intimately associated with the coupler and underframe, that it seems necessary, by way of introduction, to describe briefly the standard construction of the present day and explain the successive steps leading to its adoption.

The first means of connecting cars was by chains and hooks. This was displaced by a drawbar having an open end through which a pin passed so that cars could be connected by inserting the pins through links fitting in the opening in the drawbar. Springs were interposed between the drawbar and the car underframe to ease the shocks developed in impact and in train service. About 1880, the Janney coupler having a knuckle pivoted on a vertical pin was introduced. This type has now become standard on American railroads.

The increase in the length of trains, together with the use of the air brake, multiplied the severity of shocks to such an extent that the low-capacity spring draft gears became troublesome. Recoil after impact and after brake applications caused considerable damage to equipment, principally because the wooden underframes in use at that time developed far less strength in tension than in compression. To meet this situation, George Westinghouse developed and applied the first friction draft gear. The first patent was granted to Mr. Westinghouse in 1888, and friction draft gears came into use on a limited scale before 1900. Since that time service requirements have constantly grown more severe, and draft-gear design has become one of the major problems pertaining to freight cars. Many engineers have struggled with the problem of developing a design which would meet all requirements. The majority of these efforts have resulted only in an addition to the records of the Patent Office. There are now more than 12,000 patents on friction draft gears, but there are only about 12 different types in general use.

Present-Day Draft Gears

Draft gears being applied at the present time are all adapted for application with standard draft-gear attachments, as shown in Fig. 1. The coupler is connected by a key to the yoke which extends around the draft gear. On the underframe, opposite the ends of the yoke, are draft lugs on which the draft gear bears and through which the forces are transmitted to the car structure. The Farlow draft attachments provide an alternate construction in which the yoke is placed horizontally instead of vertically. With both arrangements any movement of the coupler in either direction tends to close the draft gear. The draft gear fits between sills, draft lugs, and yoke, in a space $24\frac{5}{8}$ in. long by $9\frac{1}{8}$ in. by $12\frac{7}{8}$ in.

The draft gear, to function properly, must be adapted to cushion impact when cars run together in switching. It should offer a yielding resistance to facilitate starting trains and should take up shocks which occur while the train is running, due to changes in grade or to the application of the brakes. It might seem that there would be little difficulty in deciding on the characteristics that would be required to insure proper action and in selecting and designing the elements that would best accomplish the object sought. As a matter of fact, there is very little agreement among designers, as will be seen later when some typical draft gears are shown. There is only one feature that is common to all of them—they all contain a spring. Efforts have been made to develop hydraulic and pneumatic draft gears, but they have been not successful. Draft gears which are now being applied range from simple spring draft gears with practically no capacity to absorb energy, through the class which has large spring capacity with a moderate amount of friction, coming finally to gears that derive practically all their capacity from friction and absorb over 90 per cent of the energy imparted to them.

In describing typical draft gears, the logical starting point seems to be the Westinghouse Type D, Fig. 2. This design, which was developed by George Westinghouse, has been in use for more than 20 years and is still being manufactured by the Westinghouse company. It is applied in the draft-gear pocket with a follower at each end. The gear has a malleable-iron friction barrel with a plurality of V-shaped ways on its interior surface. Eight composite friction segments or splines are wedged outwardly into these ways to produce frictional resistance against longitudinal movement of the splines. When a predetermined load has been applied to the friction wedge, the follower comes directly into contact with the outer ends of the splines and additional wedging is prevented. On release the friction members are started serially so that sticking will be less likely to occur.

Another gear which has been in use for a long period is the Sessions Type K, Fig. 3. This consists of a spring barrel, on one end of which is mounted a friction box containing three triangular friction members. The movement of the follower against the outer friction members causes the inner friction member to travel into the spring barrel at a more rapid rate, so that considerable resistance is set up by the spring and by the friction between the contacting surfaces.

The Cardwell draft gear, Fig. 4, embodies a unique arrangement of springs mounted outside the sills on a transverse spring rod. This gear also has triangular wedging members, the movements of which set up frictional resistance and compresses the springs.

The National draft gear, Fig. 5, has a central friction column with springs at the side. There are two V-shaped ways in which the friction members move. A movable follower adjacent to the small end of the column is arranged to wedge the shoes into the ways as the gear is closed. Brass pads are inserted in the followers where they come in contact with the shoes. The friction members are ground and are seated on inclined faces of the spring follower to facilitate release.

The Hall draft gear, Fig. 6, has a large number of alternated plates adapted to move relative to each other and set up friction under the pressure exerted by springs and wedging means. This gear gives a large friction area. It is interesting to note that alternated plates as used in this design are shown in one of the earliest Westinghouse patents.

The Bradford rocker-type draft gear, Fig. 7, has spring housings at each end with two pairs of inter-engaging rotative knuckles between the springs. The action is such that each spring is compressed between one of the housings and the adjacent faces of the rockers, the leverage increasing as the gear closes. Friction is obtained principally by the rockers rotating in the seats in the housings.

[1] Chief Engineer, Bradford Corporation. Mem. A.S.M.E.

Presented at a meeting of the Metropolitan Section of the A.S.M.E., New York, January 31, 1927.

The Waugh plate-type draft gear, Fig. 8, consists of spring-steel plates mounted between followers or separators having alternating convex and concave faces. The action is practically the same as that of a leaf spring, friction being set up between the spring-steel plates.

of travel which is most desirable. Some are in favor of approximately 2 in. travel, while others advocate making it as great as 4 in. The long travel has the advantage that it permits development of greater energy capacity with a given maximum force. Advocates of short travel contend that the necessary capacity

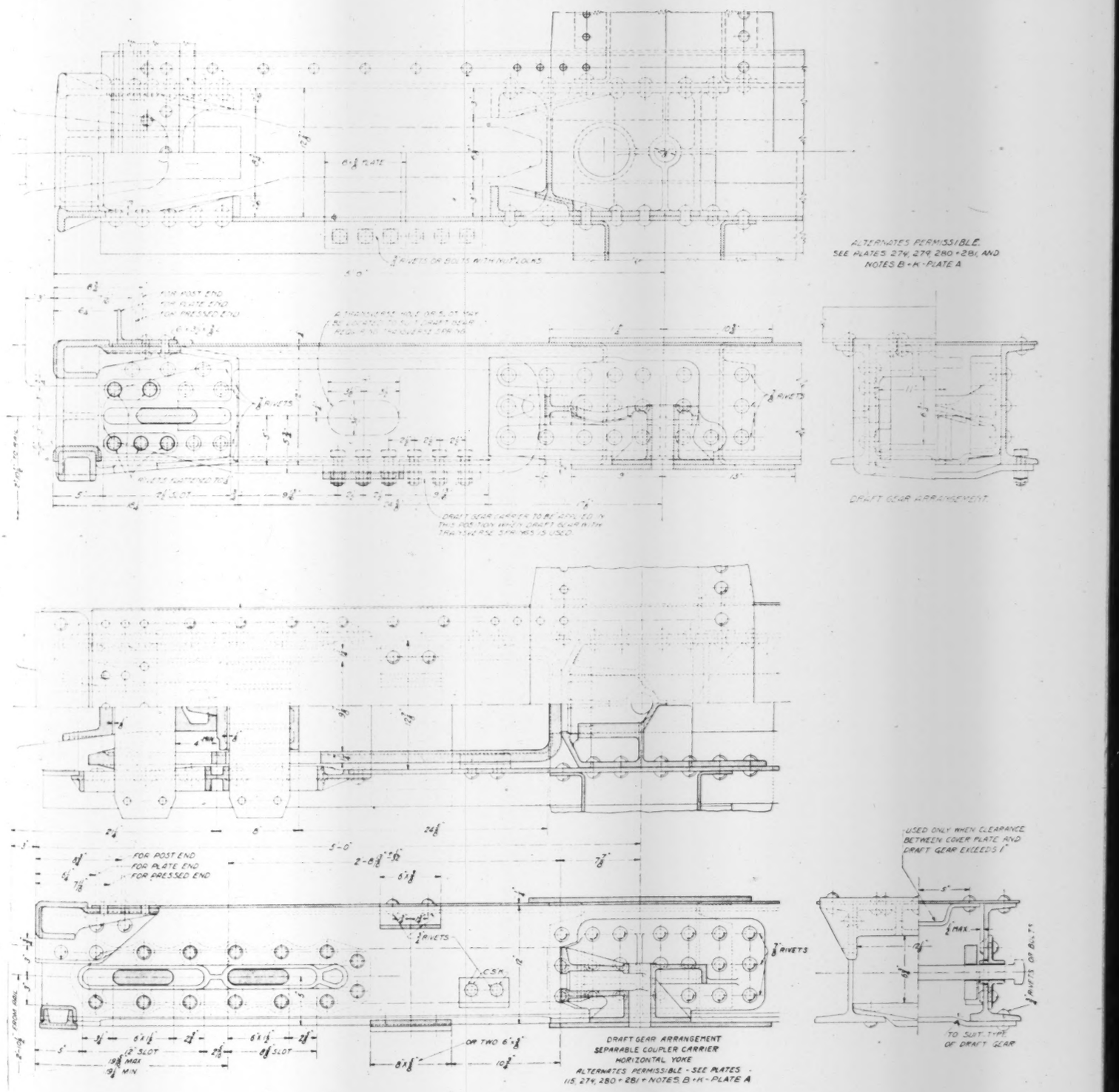
ALTERNATES PERMISSIBLE
SEE PLATES 274, 279, 280 + 281, AND NOTES B + K - PLATE A
DRAFT GEAR ARRANGEMENT
FOR POST END
FOR PLATE END
FOR PRESSED END
USED ONLY WHEN CLEARANCE BETWEEN COVER PLATE AND DRAFT GEAR EXCEEDS 1"
TO SUIT TYPE OF DRAFT GEAR
DRAFT GEAR ARRANGEMENT
SEPARABLE COUPLER CARRIER
HORIZONTAL YOKE
ALTERNATES PERMISSIBLE - SEE PLATES
115, 274, 280 + 281 + NOTES B + K - PLATE A

FIG. 1 STANDARD ARRANGEMENT OF DRAFT ATTACHMENTS WITH VERTICAL AND HORIZONTAL YOKE

CHARACTERISTICS REQUIRED IN DRAFT GEARS

At the present time there is no standard specification for draft gears, and there is little agreement as to the characteristics required. The Mechanical Division of the American Railway Association, in its Fundamentals of Car Design, specifies a minimum capacity of 150,000 lb., and a travel of 2 3/4 in. As a rule, a minimum impact capacity of about 12,000 ft-lb. is required by the railroads. Authorities are not agreed as to the amount can be obtained in 2 in. without exceeding the maximum allowable force, and that long travel is likely to permit too much movement in train service and thus set up destructive shocks.

The form of force-closure curve is subject to many modifications. The typical curve for a spring gear, of course, starts at zero and increases in proportion to the travel up to the maximum capacity of the spring. Some friction draft gears have a preliminary spring action followed by increasing resistance which finally reaches the

maximum and then remains constant to the end of the travel. Others build up resistance at a rapid rate within the first part of the travel and then increase more slowly to keep the maximum force low and still develop high energy capacity. High initial resistance is sometimes provided with the object of preventing movement under light forces as in train movement and thus decreasing wear. In others, the rate of increase of resistance is low at the start and builds up rapidly in the last part of the travel. The force developed by returning to open position also varies greatly. The work done in closing the gear which is returned during the opening movement varies from 89 per cent to 9 per cent. In many friction draft gears the return force is very low, and some of them show a tendency to stick in the closed position in service.

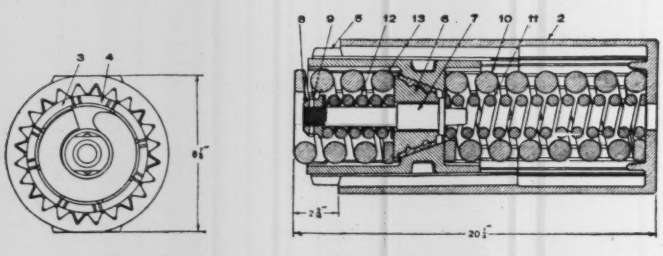

FIG. 2 WESTINGHOUSE TYPE D DRAFT GEAR

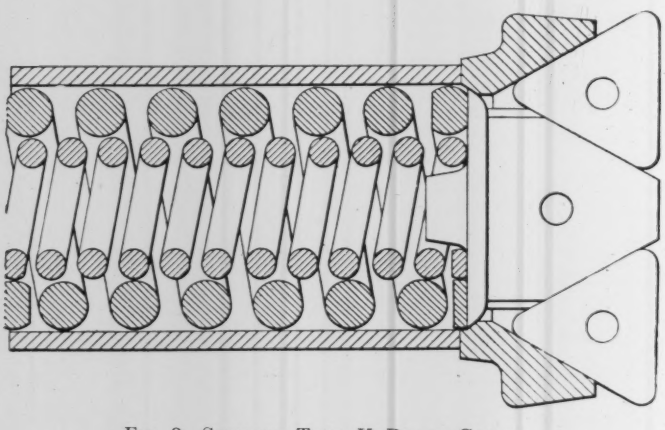

FIG. 3 SESSIONS TYPE K DRAFT GEAR

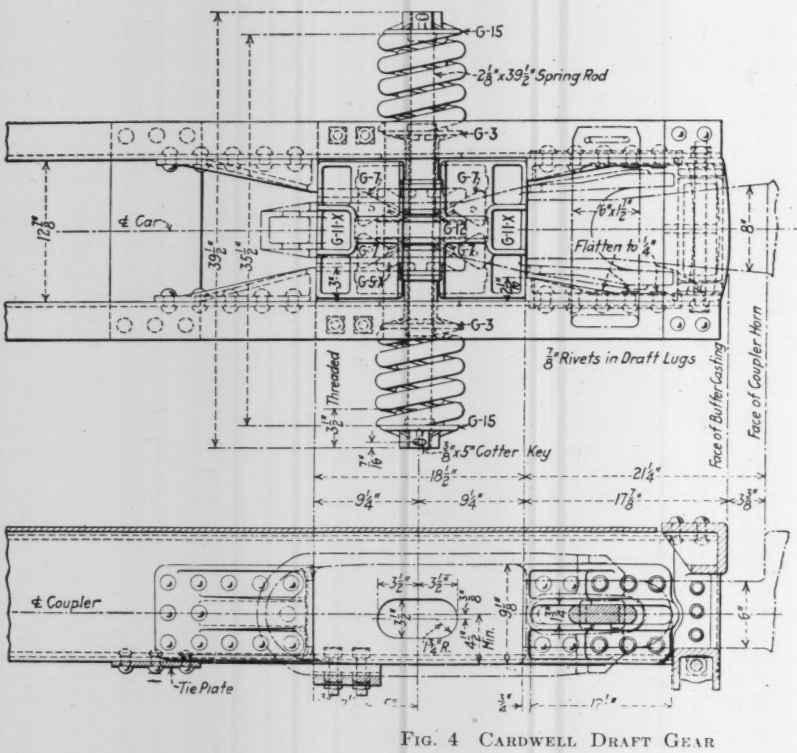

FIG. 4 CARDWELL DRAFT GEAR

Smoothness and uniformity of action are very desirable. Some gears are greatly affected by the deposit of rust and dirt on the friction surfaces and some are subject to a considerable decrease in capacity, due to moisture in the atmosphere. Some draft gears

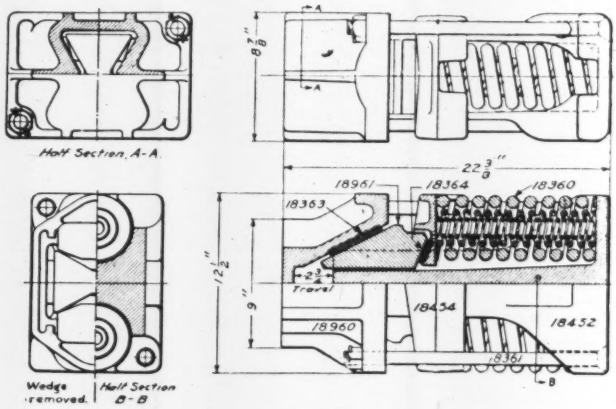

FIG. 5 NATIONAL TYPE M-17 DRAFT GEAR

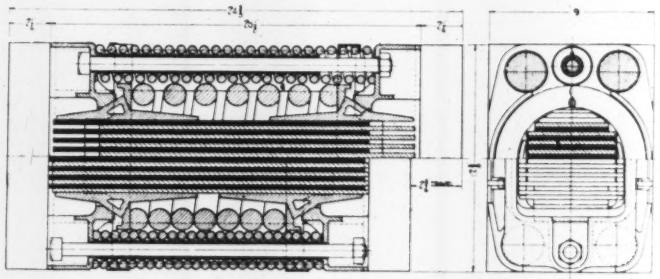

FIG. 6 HALL DRAFT GEAR

are readily closed by the repeated application and release of a force far below the capacity developed when the gear is closed in single movement. This so-called creeping is an undesirable characteristic, as the same action might take place due to the variation in the tractive force of the locomotive.

Sturdiness is a very vital characteristic because the draft gear is subjected to enormous forces in oversolid impacts. The A.R.A. recommendation is that the draft gear shall have a minimum strength equivalent to 20 sq. in. of Grade A cast steel. This is intermediate between the coupler which has a cross-section of 15 sq. in. and the center sill with an area of 28 to 30 sq. in. It seems questionable whether anything is gained by making the strength of the draft gear greatly in excess of the strength of the coupler.

It is essential that the rate of wear in a draft gear be relatively low, as it receives attention at very infrequent intervals. In some designs the pressure is made comparatively low by increasing the friction area with the object of preventing abrasion. Unit pressures in different designs vary from 200 to 5000 lb. per sq. in. on

the friction surfaces. The National draft gear illustrated has a friction area of 53 sq. in., whereas one design of the Hall gear has 2000 sq. in., or approximately 38 times as much. The Hall gear uses rolled steel plates for friction members, whereas the National gear is made of special electric-furnace cast steel, hardened to reduce the rate of wear. Another combination which is used quite extensively for the friction members is hardened high-carbon steel working against malleable iron.

METHODS OF TESTING DRAFT GEARS

In setting a standard of performance for draft gears, numerous difficulties arise. It is, of course, comparatively easy to apply gears in ordinary service and remove them after a period of years to determine how they have been functioning. This is not a very satisfactory basis for comparison, as it is impossible to tell definitely whether all the gears have been subjected to similar conditions of service. Furthermore, it is desirable to have a means for determining the performance that can be expected from a draft gear without subjecting it to a long period of service. This has led to the development of numerous methods for testing draft gears.

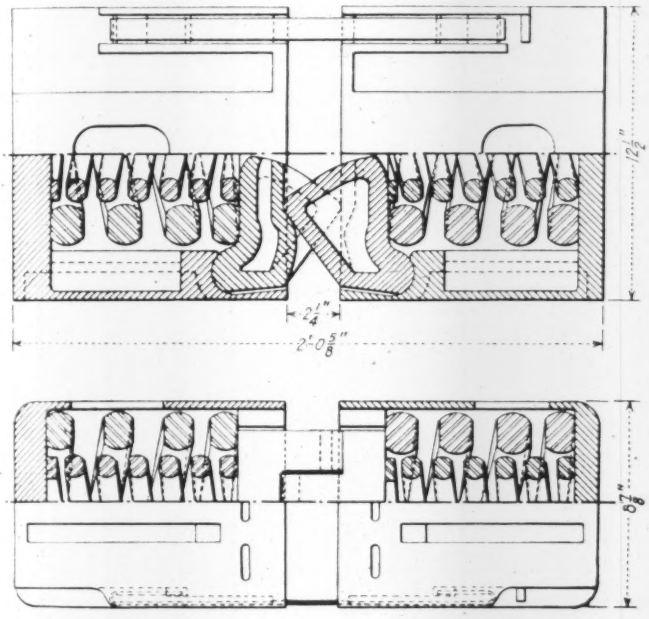

FIG. 7 BRADFORD ROCKER DRAFT GEAR

The simplest test and the one which was most used when draft gears were first developed, was the closure of the gear in a static compression testing machine. This method was satisfactory and reliable for spring draft gears, but was found practically worthless for many designs of friction gears. For example, Fig. 9 shows a comparison of the resistance developed by two draft gears of the same type under varying speeds of closure in static tests and the resistance of the same gears when closed by car impact. It will be noted that the static tests, especially at the speed of ⅛ in. per min., show great variations, whereas the capacity developed when closed on the cars corresponds fairly well for the two gears.

The realization that characteristics developed by draft gears when closed by impact are entirely different from those shown in static tests led to the development of methods for testing gears by impact. The method ordinarily used is to allow a weight of 9000 lb. to fall on the draft gear from a predetermined height. As the weight is checked by the resistance of the gear, the amount of closure is measured. The height of drop is gradually increased until the gear is driven solid. To test the sturdiness of the gear, repeated drops are made from increasing heights beyond the closing point.

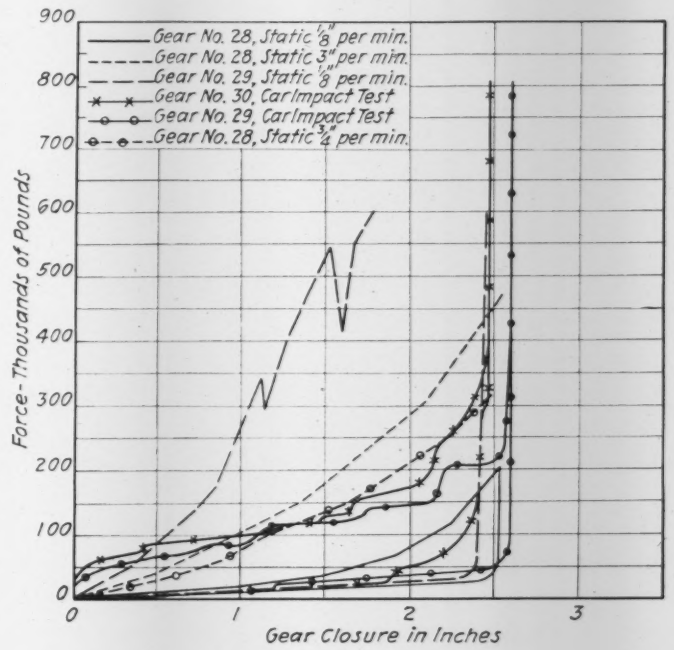

FIG. 9 FORCE-CLOSURE CURVES FOR STATIC AND CAR-IMPACT TESTS

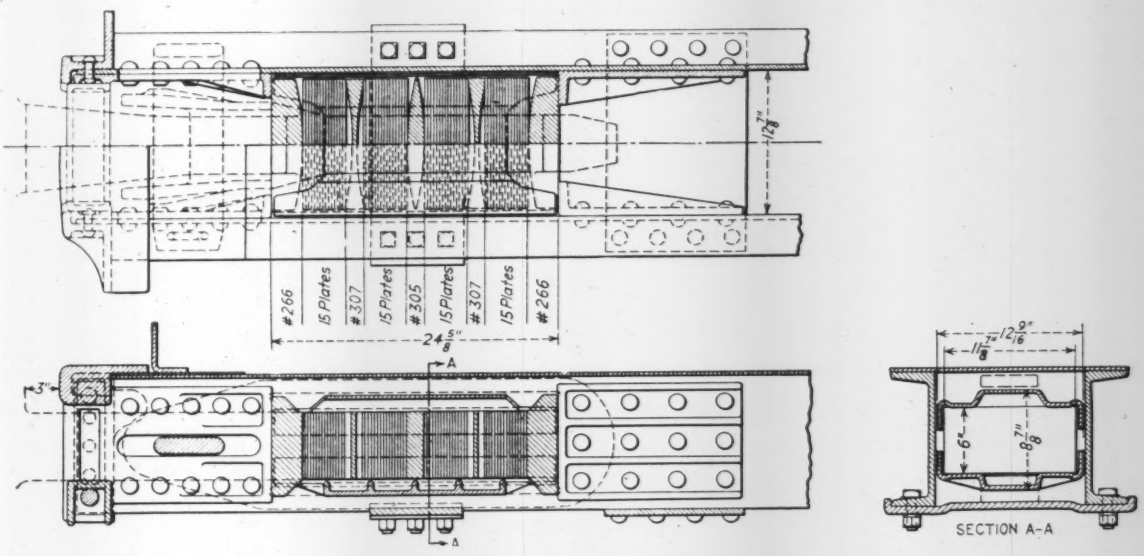

FIG. 8 WAUGH DRAFT GEAR

Various methods have been used for measuring the maximum force developed during the drop test or the force at any point in the closure. One of the simplest methods of measuring the max-

imum force is by means of a weight resting on a spring supported by the tup. The deceleration of the tup during the closure of the gear causes the weight to compress the spring. The amount of compression is recorded by a pencil, and by suitable measurements the maximum force can be calculated.

Another type of recording device is a rotating drum on which a pencil attached to the tup makes an autographic record of the movement during closure. Such a device is illustrated in Fig. 10, and a diagram is shown in Fig. 11. The slope of the line on the indicator diagram is proportional to the velocity. By ascertaining the velocity at any two points on the curve, the deceleration and the average force acting between these points can be computed. The difficulty in applying this method arises from the extremely small angles which must be measured if the forces are to be determined for points located close together.

Efforts have been made to determine the action of draft gears by the use of slow moving pictures. Very special equipment is required for this work. In a typical drop test the time of closure is only about 0.03 sec., and the time during which the gear is opening 0.05 to 0.06 sec. The ordinary slow moving picture makes an exposure each $^1/_{64}$ sec., and would take only six exposures during the closure and opening of the draft gear. Special cameras made to take 1000 exposures a second have been used to record draft-gear action. Some interesting data have been obtained by this method.

The Car-Impact Test

The test which gives the closest approach to the conditions under

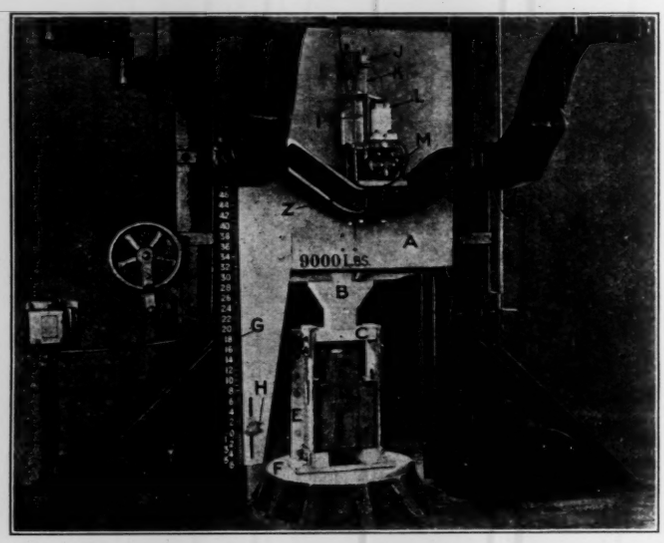

FIG. 10 RECORDING DEVICE ON DROP TEST

which draft gears operate is the car-impact test. In this test the gears are applied to cars and a moving car is allowed to strike a stationary car, complete records of the movement of both cars being obtained on recording instruments from the instant of impact until the cars part.

Before describing the car-impact test in detail, it may be well to review briefly the laws of mechanics governing direct central impact of elastic and inelastic bodies.

When two masses, m_1 and m_2, meet in direct central impact, the relation between the velocities v_1 and v_2 before, and u_1 and u_2 after collision, is represented by the equation:

$$m_1v_1 + m_2v_2 = m_1u_1 + m_2u_2$$

The sum of the momentums of the two masses before, during, and after impact remains the same. However, the kinetic energy varies during impact and may or may not be the same after impact as before.

If v = the common velocity at the instant of greatest compression,

$$v = \frac{m_1v_1 + m_2v_2}{m_1 + m_2}$$

In case the bodies are perfectly elastic, the velocities after impact become—

$$u_1 = v_1 - \frac{2m_2(v_1 - v_2)}{m_1 + m_2} \quad \text{and} \quad u_2 = v_2 + \frac{2m_1(v_1 - v_2)}{m_1 + m_2}$$

$$\text{For } v_2 = 0, \quad u_1 = \frac{(m_1 - m_2)v_1}{m_1 + m_2} \quad \text{and} \quad u_2 = \frac{2m_1v_1}{m_1 + m_2}$$

If the two bodies are perfectly inelastic, they move after impact with a common velocity

$$u = u_1 = u_2 = \frac{m_1v_1 + m_2v_2}{m_1 + m_2}$$

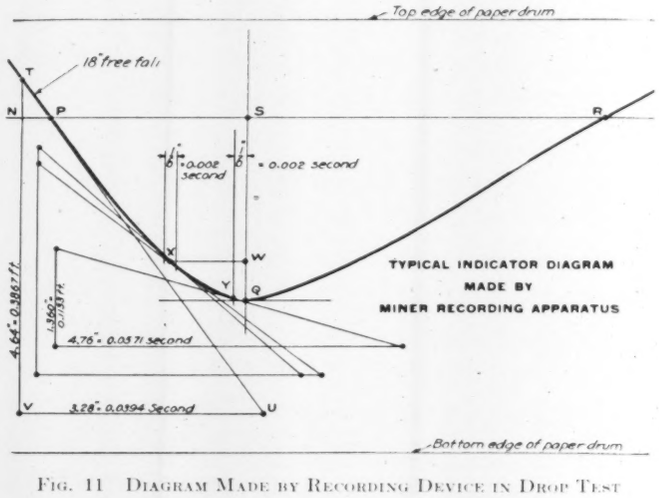

FIG. 11 DIAGRAM MADE BY RECORDING DEVICE IN DROP TEST

The loss of kinetic energy is

$$E = {}^1{}_2m_1m_2\frac{(v_1 - v_2)^2}{m_1 + m_2}$$

Collisions between cars equipped with spring draft gears approximate the condition of elastic impact. On the other hand, cars equipped with friction draft gears act more nearly in accordance with the laws of inelastic impact.

Applying the formulas given above to specific cases brings out some interesting features regarding the theoretical action of cars in impact. Assume first that a moving car strikes a standing car, both being of the same mass. If the cars are equipped with spring draft gear and the impact is not in excess of its capacity, the striking car will be slowed down and the struck car will be accelerated up to the instant of maximum compression, when both will be moving at uniform velocity equal to one-half the initial speed of the striking car. The draft gears will continue to exert force on the cars until the gears are again fully open. The deceleration of the striking car and acceleration of the struck car will continue until they part, the striking car coming to rest and the struck car moving on with a velocity equal to the initial speed of the striking car. If the striking car is one-half as heavy as the car which is struck, the striking car will move in the reverse direction with one-third its original speed and the struck car will move in the direction of the impact at two-thirds the speed of the striking car. If the striking car is twice as heavy as the car that is struck, the striking car will continue in motion at one-third its original speed and the struck car will move on in the same direction at four-thirds the original speed of the striking car. If the mass of the striking car is many times as great as that of the struck car, the velocity of the latter after impact will be nearly twice the original speed of the striking car.

In the case of inelastic impact, as between cars equipped with friction draft gears, the action up to the point of maximum compression is the same, but there is no restitution of the energy expended in compressing the draft gears and the cars move on at the common velocity of the two bodies at the time of maximum compression. In elastic impact the final decrease in the velocity of the striking body and the final increase in the velocity of the struck body are twice as great as in inelastic impact.

In actual service there is always some absorption and some restitution of energy, so that neither spring nor friction draft gears act strictly in accordance with the equations given.

It is interesting to note that if the resistance of the draft gear is proportional to the closure, the time of closure is the same for light or heavy impacts within the capacity of the gear.

The energy and force relations outlined in the preceding paragraphs are extremely important in designing draft gears. Referring again to the typical case of impact between two cars of equal weight, it is evident that the draft gear in each car should be of such construction that the work required to close it at the maximum permissible velocity of impact would be equal to one-fourth the kinetic energy in the striking car at the instant of impact. This is modified somewhat by the frictional resistance in the movement of the car and the work done on the car structure.

FIG. 12 GENERAL VIEW OF SYMINGTON TEST PLANT

Much valuable information concerning the actual performance of draft gears in impact has been obtained on the testing plant built in 1917, by The T. H. Symington Company, at Rochester, N.Y. This test plant, Fig. 12, consists of a straight track with two full-size cars which can be caused to collide at any desired velocity, means being provided to record the results of the impact (Fig. 13). The first portion of the track is inclined in order to impart velocity to one of the cars. At the foot of the incline is a section of level track where the impact takes place. The cars can be equipped with any type of draft gear and are fitted with dummy couplers so that they are free to move after impact under the influences of the forces set up during the draft-gear cycle.

During the test run, records can be taken of the velocities of the striking car and the struck car, of the travel of the cars along the track, and of draft-gear travel and action. From these records the action of both the gears and the cars can be studied.

The principal record is the time-displacement curves for the two cars. These curves are traced by pencils on two drums mounted on a common shaft and placed on a stand alongside the track. The axis of the shaft is parallel to the track and the drums are so mounted that at the instant of impact one is alongside of the striking end of the moving car, termed Car A, and the other alongside the struck end of the stationary car, termed Car B.

The draft-gear travel and action are recorded by a pencil moving on a motor-driven drum on Car B. In addition, both cars are equipped with friction-plunger gages to show the amount of coupler travel. Each car also has a pendulum device with a light friction runner which gives a record of the maximum acceleration of the car.

In making a test run, Car B is spotted at a definite location on the track. Car A is brought up until the buffing faces of the couplers are just in contact. The drums are then rotated with the pencils in position, thereby drawing the datum lines. At the same time, the small drum is rotated so that its pencil draws the datum line for the draft-gear movement. These lines represent the position at the first instant of impact and furnish the common reference point for a comparison of the records. With the drums stationary, each of the pencils is given a slight longitudinal movement to draw reference lines *D-D* and *E-E*. Car A is then drawn away from car B and the pencil for Car A is drawn along the axis of the drum in order that the approaching car may propel the pencil for some distance before it reaches the datum line. By this means the exact impact velocity of Car A is determined, the speed of rotation of the drums being known. Car A is drawn up the incline a distance sufficient to develop approximately the speed desired, and is released by a trip. As Car A approaches Car B, all of the drums are set in motion a sufficient time in advance to enable them to reach a constant speed before the pencils are moved. During the impact the pencils, propelled by the movement of the cars, draw time-displacement curves on the drums. By means of the datum and reference lines the curves can be superimposed to facilitate the analysis of the data. Such curves are shown in Fig. 14. The line starting at the lower left side and inclined upward is a curve traced by Car A. The heavy line starting at the junction of the datum line and the curve for Car A is the curve traced by Car B. The vertical distance between the superimposed curves represents draft-gear compression, together with a slight yield of the car bodies. The slope of the time-displacement curves represents the velocity of each car. It will be noted that during the first part of the impact the velocity of Car A is decreased and Car B is set in motion, the force between the cars compressing the draft gear. At the point of maximum gear closure, both cars attain the same velocity. The elastic force between the cars continues to act and the velocity of Car A is further decreased and the velocity of Car B increased, until the cars part, as indicated by the crossing of the time-displacement curves. From the car-movement curves are derived velocity curves, energy curves, time-force curves, and time-closure curves. The velocity curve, Fig. 16, is the first derivative of the time-distance or car-movement curve. The force curve or acceleration curve, Fig. 17, is the second derivative of the car-movement curve, or the first derivative of the velocity curve. These are obtained by mechanical differentiation. The energy curve, Fig. 18, is derived by calculation from the velocity curve. Two time-closure curves are obtained: one from the car-movement curves, Fig. 19, and the other directly from the small drum mounted on Car B. A comparison of these two records gives a check on the accuracy of the instruments. By combining data from the time-force curves and time-

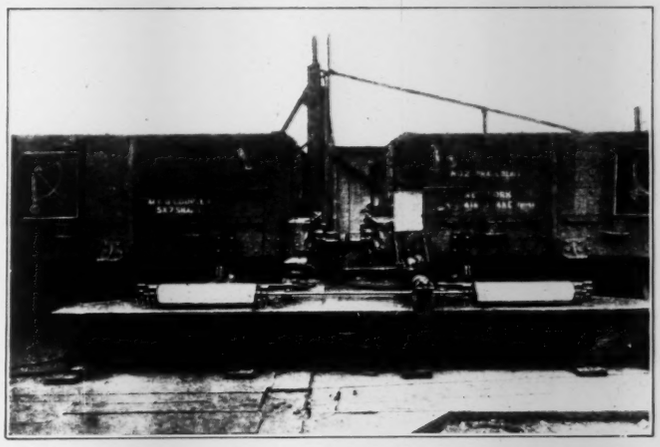

FIG. 13 INSTRUMENTS FOR RECORDING CAR ACTION

closure curves, the force-closure curves, Fig. 20, are derived. These correspond to the force-closure diagrams obtained on static testing machines, but show the true resistance developed by the draft gear when closed in impact.

The records of car movement obtained at the Symington test plant show that during impact vibrations are set up in the side members of the car. These vibrations cause irregularities in the car-movement curves which are accentuated in the velocity curve. Ordinarily, before the force curve is drawn, a smooth line must

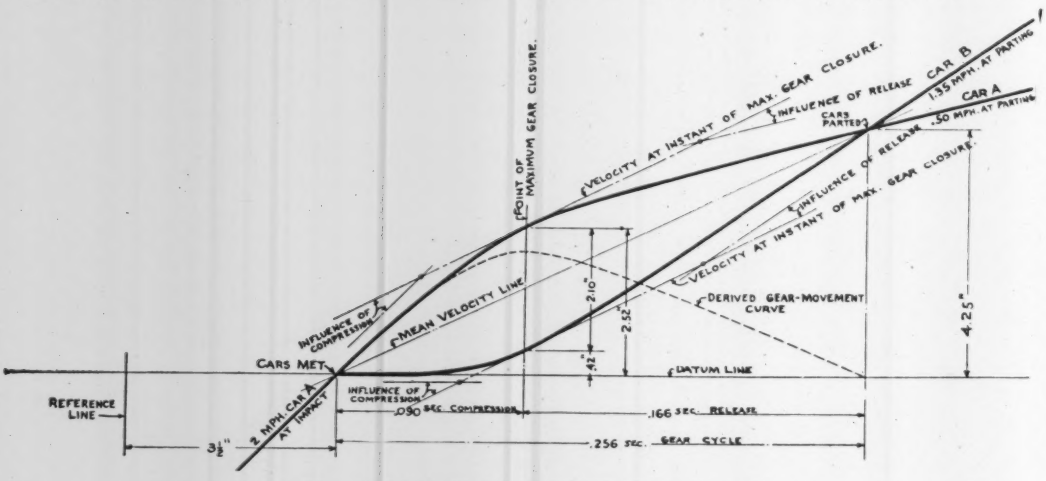

FIG. 14 CAR-MOVEMENT CURVES, SUPERIMPOSED

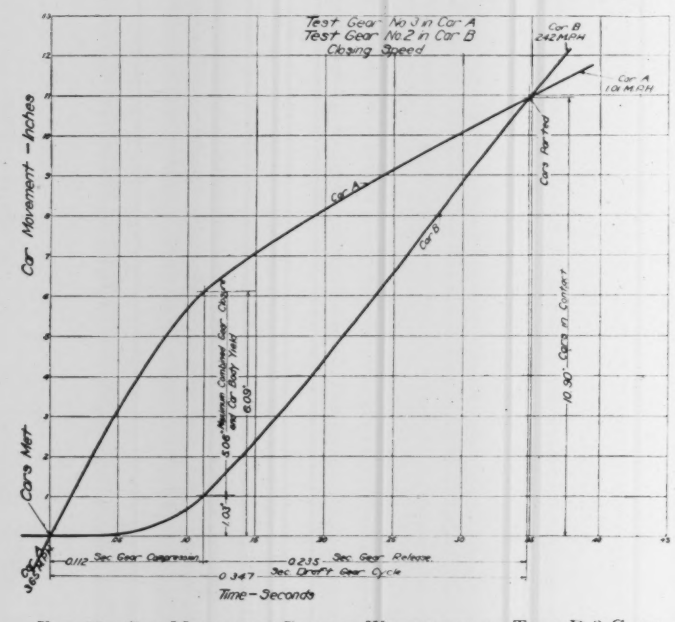

FIG. 15 CAR MOVEMENT CURVES, WESTINGHOUSE TYPE D-3 GEAR

be laid in through the mean points of the velocity curve as drawn by the differentiating machine. There is a chance of serious inaccuracy in the resulting data unless extreme care is used in redrawing the curve.

The car-impact tests prove that the action of spring and friction draft gears agree in general with the theory of elastic or inelastic impact. The results of such tests do not always correspond with those obtained in drop tests, as will be shown later.

The work done by the car body is a rather important factor in the car-impact tests. This depends on the maximum force developed by the draft gear in closing. Where the force is as great as 400,000 lb., the work of the car structure may amount to over 4000 ft-lb. In some of the U.S.R.A. tests, the work of the car structure was over 8000 ft-lb., but it is probable that where these high values were recorded the draft gears reached the limit of their travel before the velocity of the two cars became equal.

It is interesting to note the variation from the theoretical action in impact with spring and friction gears. In the test with two Class G springs, the closing speed was 1.84 miles per hour. At the end of the impact, Car A came to rest and Car B moved on with a velocity of 1.71 miles per hour. The kinetic energy in Car A at impact was 16,666 ft-lb., and the energy lost during the draft-gear cycle was 2205 ft-lb., the absorption of the gears being 11 per cent.

At the other end of the scale is the Cardwell G-18-A draft gear which showed the highest absorption, 91 per cent. The closing speed run for this gear was 3.85 miles per hour. When the cars parted after impact, Car A was moving at 1.19 miles per hour, and Car B at 2.49 miles per hour. The kinetic energy in Car A at impact was 72,930 ft-lb., and the energy lost during the draft-gear cycle was 35,476 ft-lb. The energy returned by two draft gears was only 3083 ft-lb., but the kinetic energy in the two cars at parting was 37,454 ft-lb. This shows that the car bodies store considerable energy during compression which is given back as the cars part during the opening movement of the draft-gear cycle.

The car-impact test seems the most logical means of rating draft gears on the basis of laboratory performance. In spite of this fact, it has not come into very general use. Both railroads and

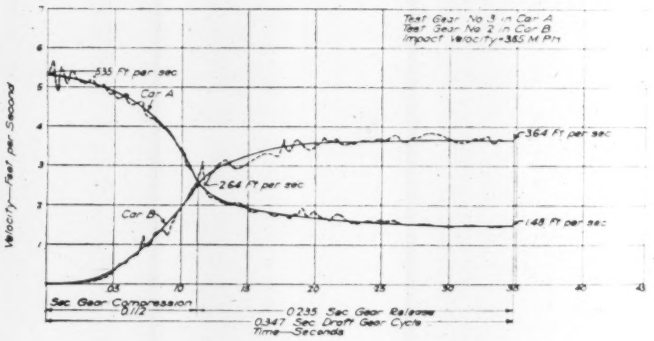
Test Gear No 3 in Car A
Test Gear No 2 in Car B
Impact Velocity = 3.85 M.P.H.
Velocity - Feet per Second
5.35 Ft per sec
Car A
3.64 Ft per sec
2.64 Ft per sec
Car B
1.48 Ft per sec
Sec Gear Compression
0.112
0.235 Sec Gear Release
0.347 Sec Draft Gear Cycle
Time - Seconds

FIG. 16 VELOCITY CURVES, WESTINGHOUSE TYPE D-3 DRAFT GEAR

(Dotted lines represent instantaneous car velocities as determined from the original car-movement curves. The irregularities are due in general to vibrations of the car structure induced by draft gear action. Full lines represent the mean velocity curves.)

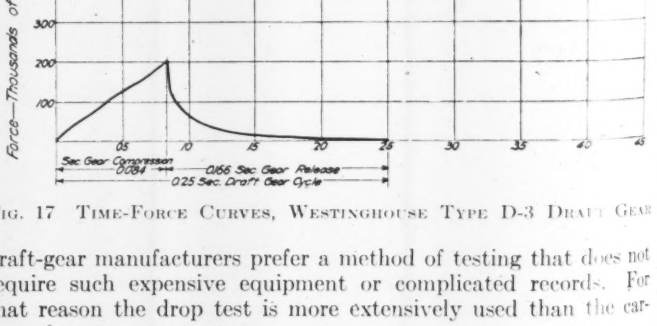
Solid Buffer in Car A
Test Gear No 2 in Car B
Impact Velocity = 2.68 M.P.H.
Force - Thousands of Lbs
Sec Gear Compression
0.084
0.166 Sec Gear Release
0.25 Sec Draft Gear Cycle

FIG. 17 TIME-FORCE CURVES, WESTINGHOUSE TYPE D-3 DRAFT GEAR

draft-gear manufacturers prefer a method of testing that does not require such expensive equipment or complicated records. For that reason the drop test is more extensively used than the car-impact test.

The velocity at impact under a 9000-lb. falling weight is considerably greater than the velocity of a loaded car which in impact would do the same amount of work on the draft gear up to the closing point. It has been suggested that because of the decrease in the coefficient of friction resulting from an increase in speed of movement, friction draft gears do not give the same performance in the drop test that they do in actual service. It is interesting to analyze the various tests conducted during the U.S.R.A. investigation of draft gears with a view to determining how the speed of closure affects the work done. In the U.S.R.A. tests, 18 dif-

ferent designs of draft gear were tested under the 9000-lb. drop, and in car impact in single-gear runs with a solid buffer in Car A and a draft gear in Car B, and in double-gear runs with two draft gears of the same type in Cars A and B. If the variations in performance are due to changes in the coefficient of friction resulting from increases or decreases in speed, it is to be expected that the work done in the drop, where the velocity was greatest, would be the least, with the double-gear run next, and the single-gear run developing the greatest capacity. A comparison of the work done in these tests is shown in Table 1. There is little evidence to indicate that speed is an important factor in draft-gear performances within the limits covered by these tests. There is little agreement between the ratio of work done in the double-gear test run to work done by the same type of gear in the drop test, the ratio ranging from 71 to 139 per cent, the average being 106 per cent. The average ratio of 106 per cent is in agreement with the theory that the work done increases as the speed decreases. Comparing the ratio of work done in the single-gear closing-speed

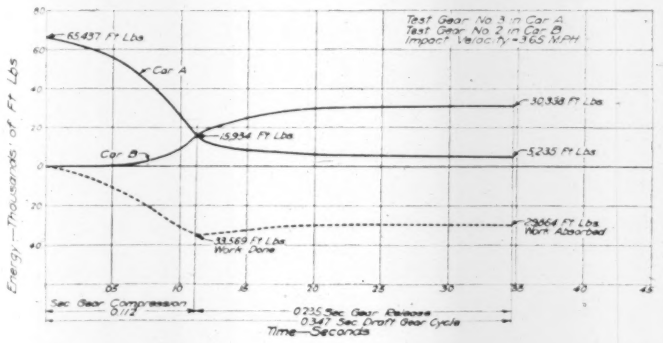

FIG. 18 ENERGY CURVES, WESTINGHOUSE TYPE D-3 DRAFT GEAR

(Full lines represent the instantaneous kinetic energy of the moving cars. Dotted lines represent the energy stored and absorbed during the draft-gear cycle.)

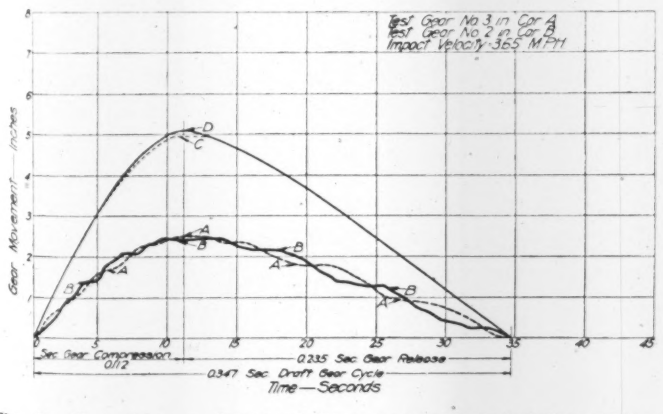
Test Gear No. 3 in Car A
Test Gear No. 2 in Car B
Impact Velocity-36.5 M.P.H.
Gear Movement—Inches
Sec. Gear Compression 0.112
0.235 Sec. Gear Release
0.347 Sec. Draft Gear Cycle
Time—Seconds

FIG. 19 TIME-CLOSURE CURVES, WESTINGHOUSE TYPE D-3 DRAFT GEAR

(Curve D, determined from superimposed car-movement curves, represents combined draft-gear movement and yield of car bodies. Curve C obtained by eliminating car-body yield from curve D, represents true combined movement of both gears. Curve B, traced on small drum, represents movement of gear in Car B. Curve A, derived from curves C and B represents simultaneous movement of gear in Car A.)

runs to the work done in the drop test, shows a wider variation, the limits being 79 to 198 per cent, with an average of 121 per cent. The ratio of work done in the single-gear run to work done in the double-gear run varies from 84 to 169 per cent, the average being 114 per cent.

Aside from the variations in the coefficient of friction, the capacity of draft gears in the drop and car-impact tests might be affected due to atmospheric conditions or the condition of the friction surfaces, by slight inaccuracies of manufacture, causing variations in capacity of different gears of the same type, by the effect of alternate closing in the double-gear runs and by variations in operation due to the position of the gear, which is placed vertically in the drop test and horizontally in car-impact tests. It seems impossible to determine separately the effect of each of these factors. The performance of draft gear is influenced by so many variables that no two tests could be depended on to give identical results.

The American Railway Association has authorized an appropriation for the purpose of building, installing, and housing a drop-test machine for testing draft gears to determine their capacity recoil, smoothness of action, sturdiness, and endurance, and from the information obtained prepare suitable specifications under which the railroads may purchase draft gears that are known to meet the prescribed standards of efficiency. It will also be used to obtain information that will be of assistance in developing draft gears generally.

A novel feature of the machine is that it will be provided with two falling weights or tups. The larger one will weigh 27,000 lb., and is believed to be heavier than any heretofore used for the purpose, while the smaller one will weigh 9000 lb., which is the weight most frequently used in similar test machines used in the past. The design is such that the weights may be readily removed or applied without dismantling the vertical columns.

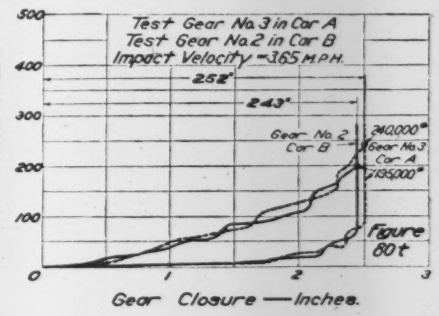

FIG. 20 FORCE-CLOSURE DIAGRAM, WESTINGHOUSE TYPE D-3 DRAFT GEAR

The machine will be driven electrically, the control equipment being so designed that operation may be manually or automatically controlled, and it will be provided with a chronograph for recording the action of the draft gear or gears being tested throughout the cycle of compression and release.

The drop-test machine is now nearing completion and will be installed at an early date at Purdue University, Lafayette, Indiana.

* * *

W. E. Symons,[2] in discussing the paper, said that following his successful introduction of air brakes on railway trains in the U. S., George Westinghouse had gone to England, where he had made much progress against what appeared to be greater opposition than he had encountered in this country.

An exhibition or demonstration of the practicability of the Westinghouse brake had been given on one of the railways outside of the City of London, known as the Galton-Westinghouse tests, which had been attended by many notable men in the fields of finance and railway operation.

On the return trip to London from this trial test, the engineer, or engine driver, in attempting to give an extra, or final, exhibition of his ability to make a quick, effective stop at the terminal station,

[2] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

TABLE 1 COMPARISON OF DROP TESTS AND CAR-IMPACT TESTS

Name of gear	Work done in drop test, ft-lb.	Work done single-gear, closing-speed runs, ft-lb.	Work done single-gear run, per cent of work done in drop test	Work done, double-gear, closing-speed runs, ft-lb.	Work done double-gear run, per cent of work done in drop test	Work done in single-gear run, per cent of work done in double-gear run
National H-1	23,400	33,633	144	27,184	116	124
Sessions K	14,100	28,017	198	19,366	137	145
Miner A-18-S	14,925	20,117	135	18,716	125	108
Westinghouse N-A-1	18,750	16,167	86	19,167	102	84
National M-1	14,400	21,300	148	20,000	139	106
Sessions Jumbo	21,075	21,317	101	19,025	90	112
National M-4	16,125	25,000	155	18,466	114	135
Cardwell G-18-A	14,800	16,500	111	17,116	115	96
Cardwell G-25-A	14,175	18,233	129	17,916	126	102
Westinghouse D-3	14,850	15,617	105	14,666	99	107
Gould 175	13,575	14,900	110	13,767	101	108
Murray H-25	12,750	14,800	116	13,900	109	106
Christy	14,700	21,817	148	12,933	88	169
Miner A-2-S	9,900	10,500	106	10,025	101	104
Waugh Plate	10,425	8,283	79	9,100	87	90
Bradford K	8,100	7,833	97	6,833	84	115
Harvey 8 x 8 springs	7,025	6,650	95	4,991	71	131
Two 8 x 8 Class G springs	4,350	...	...	4,116	95	...

had overdone the matter, with the result that the train, after coming to a state of rest, had suddenly snapped apart on release of the brakes, through the destructive action of stored-up energy in the buffing springs between the cars. This sudden kicking of the detached portion of the train in the reverse direction had thrown many of the guest passengers to the floor of the cars, obviously materially marring the pleasure of their otherwise pleasant trip, and incidentally somewhat disturbing their previously formed favorable opinion of the new air brake.

Mr. Westinghouse was much chagrined at this unfortunate incident and immediately saw the necessity for a shock-absorbing device that would so function as to preclude the possibility of such accidents and at once worked out the details of a friction gear.

During the intervening years, Mr. Symons continued, numerous imitations of, as improvements on, the friction gear had been brought out; in most of these, however, the fundamental features of a device to function as its original designer intended it should, and for which there was a crying need today, had been overlooked or ignored, the trend in many designs being only for increased capacity, without any definite feature, that would positively function under all operating conditions to prevent the dangerous kick from recoil action of compression springs which was the very thing and the most important function the friction gear had been invented, designed, and intended to perform.

Certainly we had not only strayed far from the trail blazed by the pioneer, but in Mr. Symons' opinion the majority of friction gears now in use did *not* function under all operating conditions, and their failure was in the most important feature for which the device had been originally designed, and as now used on our greatly increased-capacity freight equipment.

He ventured the assertion, although he had no definite figures at hand, that if the draft gears on our freight cars were exposed to view for ocular inspection of car inspectors as to physical condition and actual functioning in closing and release action, as clearly as the wheels, brake beams, brake shoes, axles, oil boxes, contained parts, etc., and were marked out with B.O. chalk marks for the repair track, that at least 30 per cent of revenue freight cars would at once go out of service at the very first inspection point.

This condition appeared to Mr. Symons to be the result of two things: first, a departure from the underlying, or fundamental, principles involved, and second, highly developed commercialism; and until these features were corrected, the friction-gear problems would, in all probability, remain in a more or less chaotic state.

Some gears of high resisting capacity, as shown by laboratory drop tests, deserved to be rated high as destructive agents to car and contents, and correspondingly low in the essential elements of an efficient shock-absorbing device under all actual operating conditions. This condition, Mr. Symons believed, applied to nearly all friction gears in which the friction capacity predominated over springs in total capacity and where friction elements functioned through action of angle or wedge blocks, held in contact through resistance of compression springs.

This design, he thought, was wrong in theory and defective in practice. The situation at the present time was one in which we might look for most important changes in the not-far-distant future.

Some 25 years ago, when the automatic-coupler problem had been in a state of development somewhat similar to that of the draft-gear problem at the present time, there had been patented between 4000 and 5000 car couplers, and there had been in actual use about 45 to 47 different kinds for which railways had to carry in stock repairs or replacement parts.

Most of the 5000 inventors of couplers and all of the 45 to 47 manufacturers had not only been willing to admit, but had been quite insistent as to theirs being the only device that should be allowed on equipment, while many of them had been the wildest kind of freak contraptions with more elements of danger than the old link-and-pin coupler.

In the draft-gear field today there was a great variety and assortment of friction gears—some 48 different types and priced devices on the market made by about 12 or 13 different manufacturing concerns. While the author had said that more than 12,000 patents had been taken out, from the foregoing it must be clear that before much could be said about standardization, there must first be determined, in language void of ambiguity, a standard based on capacity and efficiency in actual service accomplishing the thing that it was sought to have effectively done in car and train movement; then, by an orderly process of installation and elimination, getting down in a reasonably short period of time to the best suitable device for the service requirements.

It was Mr. Symons' belief, and had been for many years, that among the most important features of friction-gear design should be the following:

The gear should be a self-contained unit of a size to completely fill the entire space of A.R.A. gear pocket, by which plan the manufacturer would be held responsible for every piece and part of the device as applied.

Gears should be high in compression-spring capacity, and with sufficient capacity friction, entirely independent of compression springs, to not only add to total, or combined resisting capacity of the completed unit, but absolutely positive as to functioning in release movement and thus insuring absolute protection against the adverse or damaging results from kick or recoil of compression springs.

Gears should be sturdy and rugged and develop both friction and spring resistance with the slightest movement and at all points of travel, and function under all operating conditions.

There was now a wonderful field open for development, Mr. Symons concluded, and during the next five years great strides would be made. A few large companies with the best facilities for manufacture and sale, and that had developed the best-suited devices, would then enjoy the manufacturer's profits on a device that would have been simplified down to a point of close similarity in design and capacity, and to interchangeability of not only the complete unit but some of the essential parts for this work.

The Design of Propellers

WITH the growing dimensions of aeroplanes and the use of thick wing sections the efficiency of the propellers seems to reduce. In a three-engine aeroplane the center propeller seems not to develop the same efficiency as the side propellers due to the large fuselage behind it. Propellers working in a short distance in front of a thick wing lose efficiency. Notable is the fact of the big new French monoplane flying boat which they are building in France at present, with five Jupiters in the nose of the wing. The efficiency of the propellers is so bad that at present they are moving the motors three to four feet forward.

It is a well-known fact that the slow-moving propeller of a larger diameter has a higher efficiency than a fast-turning propeller of smaller size, but the extra weight, the extra cost of a gear and a larger, sometimes three- or four-bladed propeller, together with the sectional difficulty of a high-powered gear with high revolutions of light weight, are serious handicaps to its development. However, with the increased size of the aeroplanes and the number of engines it is a necessity for the future to develop larger engines with larger slow-running propellers, or to combine several engines into one unit driving on a central shaft with one single large propeller. This would permit construction of real engine rooms where better control would be possible and which would enable us to disengage defective engines and do the necessary repairs in flight.

Besides this the larger slow-running propeller decreases to a certain extent the noise, a problem with which we have to deal and which should be given more attention. A slow-running propeller eliminates the noise thirty per cent, and at present the noise is really not so bad. In the wooden propellers we have much less noise than with the metal ones. We have reduced the noise in our commercial ships to such an extent that you can almost say that you do not hear the noise of the engines; the actual noise you do hear is that of the propeller. It is a very big problem and it is worth while calling everybody's attention to it, this problem of eliminating the noise of the propeller.—Anthony Fokker in his paper on Transport Airplanes at the first National meeting of the Aeronautic Division of the A.S.M.E., Buffalo, N. Y., April 25–26, 1927.

Machinery's Contribution to Wealth and Welfare

Increase of Production Brought About Mainly Through the Successful Transfer of Skill and Intelligence from Man to Machine—Underlying Importance of Machine-Tool-Building Industry in Our Mechanized World Warrants a Larger Share of Prosperity for It Than It Now Receives

By E. F. DuBRUL,[1] CINCINNATI, OHIO

IN THE year 1776 Adam Smith, a professor of moral philosophy at Glasgow, published his enormously important work, The Wealth of Nations. So keen were the author's observations on the business system of his day that the real story of economics dates from that time; and the book is widely quoted even to the present.

In discussing the effect of the division of labor on increased production, Smith declares:

> Everybody must be sensible how much labor is facilitated and abridged by the application of proper machinery. I shall only observe, therefore, that the invention of all those machines seems to have been originally owing to the division of labor.

He also observes that a great part of the machines of his day:

> . . . were originally the invention of common workmen, who, employed in some very simple operation, naturally turned their thoughts toward finding out easier and readier methods of performing it.

Again he says:

> . . . many improvements have been made by the ingenuity of the makers of the machines, when to make them became the business of a peculiar trade; and

—Here he describes the forerunner of the modern mechanical engineer—

> . . . some by the ingenuity of those who are called philosophers or men of speculation, whose trade it is not to do anything but to observe everything; and who upon that account are often capable of combining together the powers of the most distant and dissimilar objects.

Shielded by the words of this founder of modern economics, I wish to philosophize awhile and take issue with one of the errors of the good professor, in which the present-day economists persist in following him: namely, *that division of labor is the main factor responsible for the great increase in production per man-hour.*

Transfer of Skill and Intelligence to Machines More Potent Than Division of Labor in Increasing Production per Man-Hour

The quotations I have given show that Smith did not recognize two principles that were then, and now are, more potent than division of labor in increasing production per man-hour. Eighteen years ago Dexter S. Kimball, a past-president of The American Society of Mechanical Engineers, now president of the American Engineering Council and dean of the mechanical engineering faculty of Cornell University, pointed out this great oversight. The mechanical world has been putting these principles into practice with increasing intensity for almost two centuries, yet without full, conscious realization of them.

We accomplish more with any project if we have a well-defined objective in view, a clear idea of what the principle is. Psychologists tell us that ideas are dynamic things that work of themselves. The hazy ideas quoted from Adam Smith have been at work in thousands of minds since this time. If we plant Dean Kimball's clearer and better ideas in these same minds the new ideas will do better work, because they are better. Once Kimball's principles are clearly understood they will be applied more consciously and therefore more effectively than when but dimly perceived. These important principles upon which the mechanical engineering profession rests are:

Transfer of skill and transfer of thought or intelligence from men to machines.

You have been applying these principles right along, though you may not have formulated them in that condensed form. Yet there are those who say it is brutal and inhuman to put such principles into practice.

Such indignation is pure sentimentalism, due to loose thinking. But because such views are held by a considerable number they must be reckoned with and are worth discussing at this point. Some have said that if these principles are essential to increased production, the world ought to go back to the days of hand work. They fear the world will lose craftsmanship and thereby lose too much. They do not stop to think that every time a machine is made for more foolproof or more automatic than its predecessor, the transfer of skill and intelligence to the machine relieves men from drudgery, or allows a less skilled person to step up in the ranks as operator of the new machine.

Odd Popular Notions of Our Industrial System

H. G. Wells has queer notions of our industrial system that are solemnly swallowed by a large and credulous following. According to Wells and his followers, you men who organize, plan, and direct mechanical production are chaining the worker to a machine or an assembly line. They dilate eloquently on the deadening monotony of screwing a thousand bolts, number 876, into a thousand gimcracks or gadgets passing in daily parade on a conveyor. They want to return to the good old days when men were men and could take joy and pride in their work, the expression of their own individuality.

History tells us that the good old days of handicraft were days of hard, twelve-hour muscle labor at low wages. Then a few grandees lived in less real luxury than millions now enjoy. Ask the millions who drive automobiles to go back to walking. Ask the wage slaves, chained to machines, to give up their radios and washing machines and go back to a twelve-hour stint of taking pride in their work at wages that would not buy them bungalows with bath tubs and furnaces; ask the farmer to leave his tractor and self-binder and other machinery and go back to the hoe, sickle, and flail, the implements of the handicraft era. All have become too accustomed to the products, the luxuries, of this present age, produced by mechanized industry, to prefer those idealized days of so-called craftsmanship.

Craftsmanship Increased, Not Destroyed, by Mechanized System of Production

We philosophers of production know that, far from destroying craftsmanship, the mechanized system of production has increased it, both relatively and absolutely. The new trades that have come into being because of machinery require a higher grade of mental ability than before. We have a good right to parade our actual achievements before the world. To those who will not understand what we are doing, we might murmur something about mid-Victorian ideas of the evils of the factory system, and the fear of being considered mid-Victorian will leave them gasping.

Consider the great success this method has produced for the electric-power interests. To read their propaganda one would think that power is the whole thing. It has given that industry's customers, the public regulatory authorities, and the law-making bodies an appreciation of the electrical industry that they sadly lacked before the "high-hat" campaign was inaugurated. It has increased the self-respect of the industry itself. We found the utilities parading their achievements with a decidedly superior and more than a slightly exclusive air. I want the machinery interests of the country to follow that example, to get out of overalls and into dress suits, so to speak.

Power but the Handmaid of Mechanized Industry

This effective propaganda of the power companies is stealing

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Address delivered at the Machine-Shop Practice Meeting of the Chicago Section of the A.S.M.E., Chicago, Ill., March 16, 1927. Abridged.

the spot light for a minor actor and throwing the star performer into the shadow. It is time to switch the light and let the world see that power is, after all, only an adjunct to the essential principles of transfer of skill and intelligence. The mechanical men built the power-making machines, and also the machine tools with which to make them. Furthermore, the only large market for power is in the operation of yet other machines. Until mechanical inventions gave a chance for power to go to work, easily harnessed waterfalls did no more than grind grain and raise water. The two greatest falls in the world—at Iguazu River in South America and the Zambesi River in Africa—thunder away many millions of potential horsepower, and will continue to do so until these continents develop great manufacturing industries to whose machines that power can be applied. While electric power deserves a paean of praise, a greater song of praise should be raised to the mechanical wonders to which power is but as the handmaid to the mistress.

Examples of Skill Transfer

Dean Kimball cites a drilling jig as giving a very good illustration of the practice of both the principles he formulated. He says;

> To drill four holes in a plate so that they bear a certain fixed relation to each other and to the edges of the plate, and to do this with any fair degree of accuracy, requires a high degree of skill on the part of the man doing this job. But to drill any number of such plates so that the spacing of the holes in them will correspond closely with those in the first plate requires an extremely high degree of manual skill, considerable time per plate, and is a costly operation.
>
> Suppose, however, that a skilled workman makes a drilling jig in which a plate can be securely clamped, and in which all the plates in turn can be clamped in the same position. With that jig almost any unskilled person can drill the plate as accurately as the most skilled workman can without it. Not only that, but if the jig is accurately made he cannot drill the plate inaccurately. So the accuracy of the work no longer depends on the skill of the machine operator, but on the accuracy of the jig to which the tool maker's skill has been transferred. This jig, in and of itself, has nothing to do with division of labor, although by means of the jig the application of the principle of division of labor can be greatly extended.
>
> In drilling the plate without a jig the skilled mechanic must expend thought as well as skill in properly locating the holes. But with the jig, the unskilled operator need not have a thought on that point, that mental act having been done once for all by the tool maker.

Consider also a machine, installed by the U. S. Coast Survey at Washington, that calculates the tide tables for a year for each main point on our coasts. This calculation involves quite a number of variables affecting the time and height of tide each day at each place. It used to take quite a staff to do this work. Now the machine does it. The variables for a given place are set in the machine, the machine goes through the calculations, the attendant merely taking off the readings. The machine will soon be equipped with automatic recording devices, leaving the attendant free. It would be very hard to convince the staff who issue the tide tables that they are chained to that machine, or that they ought to be chained to a pencil and paper instead.

Or consider a machine recently completed at the University of Wisconsin for calculating statistical correlations. It takes ribbons of paper punched for the series of data to be correlated. The operator sets these ribbons, starts the machine, locks the room and goes off. The machine goes through some exceedingly long and intricate series of mathematical operations, of course faster than any human being can. It records every operation as performed. It stops automatically when it finishes the job set, and is ready for the next. It makes no mistakes. Now, I ask you, just whom to pity as being enslaved or chained to that machine? Has it not rather released brains from mental drudgery to the more important job of finding work for the machine to do or interpreting the results obtained?

What Modern Industrial Principles Have Done for Our Country

Here is what these principles have done for our country: In 1812 agriculture was not mechanized. The population of the country, 7,700,000 people, was mostly engaged in agriculture, digging a rather poor living out of the earth by sheer muscle power. There was neither much crop surplus to exchange for manufactured goods, nor, until Cyrus McCormick's machinery came on the market about 30 years later, was there much surplus labor available for factory work. Some factories used textile machines, and some iron works and machine shops made the few machines so far developed; but the per capita value of the manufactures was but $22 a year, or a total of $17,000,000.

By 1919 we had 196,000 factories employing almost eleven million people, or more than the entire population of 1812. The value of the manufactured products has grown to $62,000,000,000 or nearly $600 per capita, a tremendous advance even with due allowance for the unreliability of dollars as a measure for different periods. The fifty-year period from 1870 to 1920 presents a more reliable picture because we can reduce dollars to equivalent units of purchasing power. By the use of census figures reduced to indexes, with the year 1870 as the base year in each case, we can compare the relative conditions of different factors as shown in Table 1.

TABLE 1 POPULATION, EMPLOYMENT, AND VALUE OF MANUFACTURES DURING PERIOD 1870–1920 COMPARED WITH FIGURE FOR 1870 AS BASE YEAR

	Year					
	1870	1880	1890	1900	1910	1920
Population	100	130	163	197	238	274
Persons engaged in agriculture	100	130	144	175	214	185
Persons engaged in manufacturing	100	139	208	216	436	523
Equivalent value of manufactures	100	166	296	356	510	690

This table shows that while population increased 174 per cent, the number of persons engaged in agriculture increased only 85 per cent. Today, seven years later, with at least 10,000,000 more people to feed and a greater proportion engaged in factory work, the farmer finds that, by means of his machinery, he has overproduced some crops to a disastrous extent. He is crying for relief. While between 1870 and 1920 the number of persons engaged in manufacturing increased about five and a quarter times, the value of their products increased nearly seven times.

Without question the biggest single factor in this progress has been the advance in design of machinery. In this the machine-tool group, the largest of any single group of industrial-machinery producers, have kept full pace with colleagues in other mechanical lines. A recent exposition showed that all designs of machine tools even as recent as 1920 are definitely obsolete today. At the industry's own exposition, to be held in Cleveland this fall, further advances in many designs will be exhibited. A man interested in all machine tools will not be able to cover that show completely in a week.

The Machine-Tool Industry's Dollar Worthy of Its Hire

In our mechanized world, with even warfare mechanized, machine tools produce all metal consumer goods, all other productive machines, and the machine tools themselves. A healthy, progressive machine-tool industry therefore is essential for industrial progress, and vitally necessary for the national defense.

Engineering progress entails heavy costs of development and heavy risks of obsolescence that must be paid by the consumer if the industry is to continue. The machine-tool builders have the conviction that they are not sharing ratably in the prosperity they are helping their customers to attain. This is confirmed not only by comparison of their own operating statements and balance sheets with those of their customers, but by comparison of two statistical indexes, that of bank debits outside of New York and that of the values of machine-tool orders.

To get an idea of the relative behavior of these two factors we take a common base period in both cases, the four years, 1919 to 1922, inclusive, two years of boom and two years of depression and readjustment for all industries. By reducing each index for each year to a percentage of its own base we get the result shown in Table 2.

TABLE 2 COMPARISON OF MACHINE-TOOL ORDERS WITH GENERAL BUSINESS, THE AVERAGE OF THE YEARS 1919–1922 BEING 100

Year	Bank debits, per cent	Machine tools, per cent	Bank debits in per cent of machine tools
1923	108	93	116
1924	111	65	171
1925	124	104	119
1926	128	113	113
Four years' average	118	94	126

Here we see that business in general averaged 26 per cent better than the machine-tool-dollar volume. Yet the machine tools

are both more costly and more productive than those of the base period. Had the machine-tool builder's prices reflected this added cost, his index would have been in line with the bank debits at least. Had his prices reflected the added productivity, his index would have far exceeded the bank-debit index, because his new tools have increased more than 26 per cent in productivity.

Yet there has been heavy mortality of machine-tool concerns since 1920, apparently due to uneconomic underpricing. There are many shop owners who would be glad to sell out. Since there is little prospect of selling machine tools in greater quantities, the only remedy for present conditions is for the builders to get prices for their tools that at least conform to their costs. In setting these prices they must consider that the greater the productivity of a machine, the fewer the user needs. Underpricing can only be cured when the machine-tool builder decides that his labor is worthy of a better hire than he is getting.

This same condition, I believe, exists throughout the entire industrial-machinery business. The world wants all sorts of consumer goods continually and in increasing amounts, but its enlarged wants will be supplied only because some men have the ability to transfer more and more skill and intelligence from men to machines. The sooner these men realize their proper place in the general scheme of production, the sooner they will insist on and collect more nearly the hire of which their labor is truly worthy.

The Stabilization of Long-Distance Power-Transmission Systems

Particulars of a New Scheme That Greatly Enlarges the Field of Long-Distance Electric-Power Transmission and Opens for Economic Development New Sources of Power Such as Remote Waterfalls or Beds of Cheap, Low-Grade Fuels

IF POWER could be generated with equal ease and cheapness at any place in the country and in any, even very small, volumes there would be no question of power transmission. Each consumer would then have his own power plant—the situation that existed not so long ago in the industrial application of electricity. Economy of power production, however, requires certain concentration: the thermal efficiency of large prime-mover units is higher; the attendance required by a turbo-generator of 100,000 kva. is approximately the same as required by a 1000-kva. machine; the initial cost per kilovolt-ampere of capacity decreases with increased capacity of machines. All these factors decrease the cost of the generated unit of power. Concentration of power production and its distribution to many customers require that this power should be produced in the form of electrical energy, and therefore electrical generators and transmission lines appear as intervening factors between the prime mover and consumer. To decrease the cost of transmission the power houses usually were built close to the center of the load.

It so happens, however, that some places are far more suitable for generating power than others. Waterfalls are not met with where convenience in power transmission would require them to be, but where nature has placed them. Even steam power plants of modern large size usually are located at places where cheap fuel is available and where the convenience in fuel delivery is coincident with the availability of large volumes of water for the operation of the steam condensers.

Depending upon economic considerations, it is found profitable in many cases to produce power at waterfall sites or where cheap, low-grade fuels are located—sites and locations often separated from the load by a distance of 200 or 300 miles. Transmission of large blocks of power over long distances is economically possible by using high-voltage alternating current on the transmission lines. The increased length of transmission lines and the increased voltages give prominence to certain factors which could be neglected in the former small systems. Although the science and art of power transmission by electricity have already achieved a very high state of development, by no means may the problems be considered as solved, and in fact, as the rate of power consumption rapidly increases with the industrialization of the country, ever-new problems appear in the production and transportation of the power.

Electrical engineers have given the most serious attention to these problems, which happen to be of a very complicated nature and which involve many considerations in the design of the electrical generating apparatus, the transmission line proper with all its appendages, and the motors and different power-consuming devices at the receiving end of the system. Mechanical engineers are interested not only in the broad phases of the problem, but in many details as well. If additional load is suddenly applied to the shaft of a large synchronous motor at the receiving end of the system, the input into the prime mover, due to the sluggishness of the governor, cannot increase at the same instant, and the temporary deficiency will be supplied by the stored kinetic energy of the rotating masses of the motor and the generators. They will be retarded and the frequency of the system will decrease together with changes of voltages and currents all over the system until, after three or four seconds, the governor adjusts the input into the prime mover to correspond to the new load conditions and, by accelerating the turbo-generator, brings the frequency back to normal and establishes the angular position of the rotor corresponding to the new output of the generator. The voltages and currents then will also correspond to the new situation. A very important problem before mechanical engineers in connection with the stability of transmission systems is the creation of more sensitive governors that will respond, not to the change in the number of revolutions per minute, but to the accelerations and decelerations. The modern transmission system has several synchronous generators at the sending end, and a considerable part of the load at the receiving end consists also of synchronous machines as, for example, synchronous motors, synchronous converters for railway work, and synchronous condensers for improving the power factor of the load and the voltage regulation at the receiving end; there may be also an auxiliary stand-by power plant. The synchronous generators at the sending end and the synchronous machines at the receiving end may transmit the electric energy and convert it into mechanical energy on the shaft only if they are in synchronous, that is, if they are operating in step. If, as a result of some disturbance—as for instance a flashover, a phase-to-phase short-circuit, switching out a part of the circuit, or loss of a large percentage of generating capacity—synchronism is lost, the system cannot transmit any power and it becomes unstable and inoperative.

TRANSMISSION STABILITY

The following is quoted from a paper by C. L. Fortescue, on transmission stability, which appeared in the September, 1925, issue of the A.I.E.E. *Journal* (vol. 44, p. 952).

> The attention of the engineering world has been drawn to the phenomena of instability as a result of actual experience in operation. The early cases naturally had to do with links between generating stations in distribution systems. I can recall some early cases of instability due to short-circuits in feeders connecting large generating stations which were the cause of a good deal of speculation.
>
> More recently there has been a tendency to connect large utilities together with a view to interchanging power in case of emergencies. Unfortunately these tie lines have been frequently of the type very aptly defined as "shoestrings" and have failed to carry out the purpose for which they were intended. In other words, these shoestrings in themselves were unstable and would not tie the systems together. In many of these cases, fortunately, the reason for the tie was more sentimental than real, and the

systems tied together being self-sufficient, no harm was done when they broke apart. The case of a superpower interconnection would be another story, and a break apart might be serious.

We first became interested in this question as a result of a study of the possibilities of troubles due to connecting large hydroelectric generating plants through long transmission lines with large public utilities distributing power. This study has thrown a new light on this whole class of operating phenomena, and a number of happenings which were in the past mystifying in the light of knowledge gained by this study have very simple explanations.

To cite a few examples, we have all heard of the fading out of power from a hydroelectric source supplying a utility. We have also heard of power surges, which appear to be of more or less harmonic character; that is to say, a generating plant has a period during which it alternately acts as a motor or generator. These cyclic changes may be relatively slow and will be indicated by the wattmeter, which will swing in synchronism with the load changes. The system in such a case has already passed the point of stability, although not infrequently the generator ultimately falls in step again. The effect on a large system may be quite serious as synchronous motors may be thrown out of step during the period of unstable operation.

The increased lengths of transmission lines, the higher voltages used, the greater spacing between conductors, and the larger diameters of conductors increase very considerably the capacity and the reactance of the lines. From the stability point of view, resistance and capacity are not so important, but reactance is a factor of primary importance. When alternating current is flowing through a conductor it produces along the conductor an alternating magnetic field, which in turn produces in the conductor a back-electromotive force which is known as "reactance voltage drop" in the conductor. The electromagnetic energy stored in this field during one-half of the cycle is returned to the circuit during the second half of the same cycle, and therefore sustaining this alternating field does not require energy. But it requires a magnetizing current which lags 90 deg. behind the applied electromotive force and which is called "lagging reactive current." Increase of this current in the system increases the lagging component of current in the stators of synchronous machines, and the resulting armature reaction tends to decrease the air-gap flux and, correspondingly, the terminal voltage. The amount of power, however, that can be transmitted over a certain system is proportional to the square of the voltage, and therefore a voltage drop means a decrease of the power that can be transmitted and eventual loss of synchronism in the system. If a fault to ground or a phase-to-phase short-circuit occurs, a current which is preponderately lagging results, and if we do not want the generators to supply it to the system and thereby drop their terminal voltage, we must provide some sources on the system for supplying this reactive current and thereby maintaining the voltage all along the system. The means for this purpose in the system devised by Frank G. Baum are synchronous condensers, distributed along the transmission line, which supply reactive current at each point commensurate with the requirements of the system at any given moment. To be effective such synchronous condensers must be supplied with automatic voltage regulators and quick-response exciting systems so that, in case of a disturbance, the voltage-regulating equipment responds in a fraction of a second.

On the other hand, if the load is suddenly dropped, the capacity of the straight line, that is, its charging (leading) current, will produce excessively high voltages in the system. Again, in this case, the synchronous condensers distributed along the line, by quick readjustment of their excitation, will limit these excessive voltages.

FLOW IN ELECTRIC TRANSMISSION LINES AND PIPE LINES COMPARED

Substantially, the electric transmission line operates qualitatively in ways similar to a pipe carrying water under considerable pressure. The main difference between the two lies in the fact that the velocity of propagation of electric waves in a transmission line is of the order of 150,000 miles per second. Because of this, any readjustment in the line takes place at an enormously faster rate in the case of electric "flow" than in the case of water flow.

Assume now that water is being sent through a pipe from which a number of branches are tapped, some of them open and some closed. Assume next that suddenly one of these branches of comparatively large size which has been closed, is opened. It is obvious that pressure in the pipe will drop just as it does in an electric transmission line during a fault to ground. Assume, however, that at certain points in this pipe line large reservoirs are connected to the pipe line, and that when the large branch line is opened, valves in all the side reservoirs are also opened and thus cover the deficiency of water delivery by the main pump at the sending end of the pipe line. In this way the pressure in the line will be sustained. Again, assume that this large branch is closed as suddenly as it was opened. Valves into the reservoirs then open and the normal pressure will be maintained. The opening and the closing of the valves, to produce the best effect, must be as quick as possible. It is comparatively easy to visualize how stable conditions would be reëstablished in such a water line, and to all practical purposes this is substantially what is effected by Baum's synchronous condensers with quick-response excitation—the arrangement being equivalent to the above-mentioned reservoirs. Synchronous condensers formerly were used at the receiving end of the transmission system for power-factor correction and for voltage regulation at the center of the load, and such condensers must be regarded as part of the load. Synchronous condensers as proposed by Mr. Baum and distributed along a long-distance transmission line must be regarded as part of the line, changing the characteristics of the line in the direction favorable for the transmission of energy. The following is quoted verbatim from page 7 of the Baum patent (United States No. 1,617,007, Feb. 8, 1927):

> It will be seen that my system of transmission automatically regulates the operation of the uniformly spaced synchronous condensers to maintain a substantially constant voltage throughout the length of the line by supplying the wattless power taken by the line itself. In this way the reactance of the line is substantially eliminated, and the transmission losses are limited to resistance and leakage losses only. By providing the relatively short sections connected in series relation, the line inherently possesses stable characteristics under all operating conditions, and, while the voltage is maintained substantially constant, the difference in the power factors between the generating and the receiving ends is materially reduced over prior-art transmission systems.
>
> Furthermore, the sections of the transmission line constitute but a small fraction of the wave length corresponding to the frequency of the transmission system, which in the present case may be asumed as 60 cycles, although the total length of the system may be an appreciable fraction of a wave length or may even exceed a wave length. It will be understood that, in case a relatively high frequency were employed, my invention would be applicable to a much shorter transmission line than the proposed transcontinental line is to employ, such, for example, at a relatively short branch thereof. The same effects and results would be obtained in this high-frequency system and a corresponding reduction in the spacing of the synchronous condenser substations would be required. In other words, the fact that the total length of the line is an appreciable fraction of the wave length corresponding to the normal frequency of the line, is an important feature of the present invention.
>
> It will be seen that such a national power system will make the best possible use of the potential power sources of the country, because the system will be so large as to make best use of the varying stream characteristics in the various sections. Such a system will also derive advantage from the time differences or shifts in daylight hours between the East and the West by reason of the differences in time of the peak loads on the system. It will also take best advantage of the diversity of interests in the various sections of the country.
>
> Such a system of electric power transmission is necessary for the economic development of this country. For, aside from the economics of such a system in connecting up the surplus power sources with the large consuming markets and in bringing into use many power sources otherwise not feasible of development, such a system will bring about a distribution of the industries and population of the country not otherwise possible, resulting in adding to the general stability of the country.

The application of the Baum patent considerably enlarges the field of the long-distance transmission of electric power. By permitting more power to be transmitted over the same circuit, or the same power over a longer distance, it opens for economic development new sources of power, such as remote waterfalls or beds of cheap, low-grade fuels.

During the process of cleaning up the old plant of the Llanglaagte Gold Mine, the steel extractor boxes, some of which had been in use for over twenty-five years, were scaled after being out of commission. The return was good, so they were painted first with sal ammoniac and afterward with strong hydrochloric acid, and allowed to stand for a fortnight, when they were again scaled. The results were so encouraging, says the *S. A. Mining and Engineering Journal* that the process was carried on for twelve months. The total recovery from this source was 1450 oz. from twenty-three boxes. The absorption of gold in the steel is not easily accounted for, as the returns from the treatment of similar boxes in other mines varies greatly.—*The Engineer*, May 13, 1927, p. 623.

Steam-Station Development in the Middle West

A Symposium Presented at the A.S.M.E. Regional Meeting, Kansas City, April 4–6, 1927

Characteristics of High-Pressure Power-Station Design in the Middle West

By F. S. COLLINGS,[1] CHICAGO, ILL.

IN PRESENTING a brief outline of the more outstanding points of interest associated with the design of the modern power station in the Middle West, and as an introduction to the subject, it may be well first to consider the principal factors involved. It is assumed that there is available a suitable site, the necessary railroad facilities, an adequate supply of circulating water for the condensers, fuel, and a source of reasonably good water for boiler-feed make-up.

The capacity of each of the main units will depend upon existing station load, if any; the anticipated load, load factor, capacity factor over a period of, say, three years, and the planned ultimate capacity. It may be 50,000 kw. as at Waukegan and Stanton; from 70,000 to 90,000, as in the later units at Crawford Avenue; 165,000 kw. as in the proposed extension of the Philo plant, or 206,000 as in unit No. 1 at State Line.

The next major question arising is that of steam pressure, and a brief résumé of past history will explain why 500 to 600 lb. per sq. in. has become more or less standard pressure at this time for the larger stations.

Toward the end of the year 1922, that is, before the 550-lb. Philo station became an accomplished fact, it fell to the lot of the author to make a thorough investigation into economies of high-pressure steam as applied to central stations.

At the outset, the discovery was made that the authorities on the properties of steam had failed to keep in step with the progress of the engineering art, and therefore there was available neither steam table nor temperature-entropy diagram covering the pressure and temperature field under consideration.

It was finally decided to extend the existing tables to 1200 lb. absolute and to 750 deg. fahr. total temperature, using the formulations of Prof. H. L. Callendar, and it was on this basis that our subsequent calculations were made. The net result of these calculations is shown in Figs. 1 to 6, and in the final analysis in which investment and operating charges are balanced against thermal efficiency it is apparent that a working pressure of 600 lb. per sq. in. offers the best return on the invested dollar at this time. This is true even of the direct-expansion turbine, while the additional gain due to reheating at, or slightly above, that point in the expansion curve which approaches saturation at the normal expected load, is from 4½ to 6 per cent, according to the turbine characteristics.

It has been a matter of surprise to the pioneers in the reheat field that the interstage reheater justifies its investment charges over a very wide unit-load range. In addition to the greatly increased temperature and total heat head at the inlet to the low-pressure cylinder, reheating transfers the drip point from what would be a relatively high stage at fractional load to approximately the last wheel, which contributes in no small degree to low turbine depreciation as well as to increased stage efficiency.

That the regenerative cycle will be used for feedwater heating may be taken as a matter of course, but in considering the steam-extraction characteristics of the main unit, the obvious question is: At what temperature may the air for combustion be admitted to the stoker or the pulverized-fuel equipment, as the case may be? If it is to be limited to, say, 300 deg. fahr. in order to avoid undue strain and deterioration in a chain-grate stoker, then a water economizer followed by a relatively small air preheater is indicated. In such a case it is advisable to limit the regenerative feedwater heating to a value which will insure an adequate return from the economizers. As an alternative, the gas-heated water economizer may be omitted; an air heater of sufficient surface to reduce the boiler outlet gas temperature to the desired stack temperature, installed; and a fin-tube air cooler placed in the air outlet duct from the air heater in order to temper the air to the stoker requirements. This air cooler uses boiler feedwater at a fairly low-temperature point in the cycle as a circulating medium, and when arranged with a suitable bypass affords a very convenient and flexible method of regulating the temperature of the air to the stoker. This arrangement also has the advantage that the air-cooler surfaces are always clean, which is not the case with the flue-gas-heated economizer, and it escapes the corrosion due to the precipitation of acid soot, which occurs in the latter when on bank or whenever the tubes are chilled below the dewpoint either by low feedwater temperatures or by operating at low ratings.

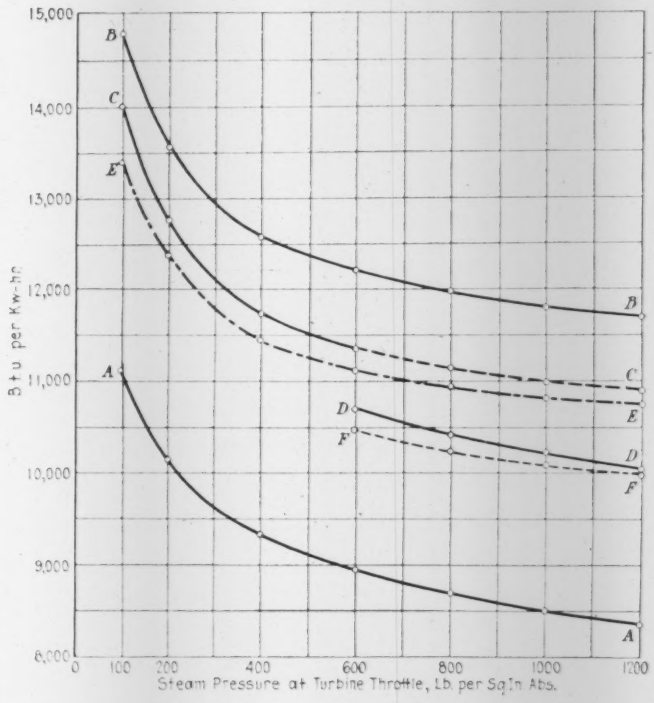

FIG. 1 B.T.U. PER KW-HR. TO TURBINE UNITS WITH DIFFERENT STEAM PRESSURES AND CONSTANT TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR. AT THE THROTTLE

A-A: Based on adiabatic expansion to 1 in. Hg abs. back pressure.
B-B: Based on actual expansion to 1 in. Hg abs. back pressure without interstage reheating or steam extraction.
C-C: As B-B, but with steam extraction sufficient to heat condensate from 80 deg. fahr. to 210 deg. fahr.
D-D: As C-C, but with interstage reheating. Steam is extracted in all cases at a point equivalent to one-fifth of the throttle pressure, and reheated to 700 deg. fahr.
E-E: As C-C, but with steam extraction sufficient to heat condensate from 80 deg. fahr. to within 60 deg. of the temperature due to boiler pressure.
F-F: As E-E, but with interstage reheating.

The new Exeter Power Station of the American Gas & Electric Company is equipped with these secondary economizers in connection with large-capacity Ljungström air heaters, and there is every indication that the installation will work out in a very satisfactory way. If on the other hand pulverized fuel is to be used, the secondary air at least may be preheated to the limit established by the gas temperatures from the boiler, qualified by the economic restriction which imposes a reasonable terminal difference. In this event the feedwater may be heated by steam extraction from the main turbine to within, say, 50 to 100 deg. fahr. of the boiling point due to the pressure. In this connection it should be noted that the reheat turbine lends itself particularly well to this latter cycle, for the reason that steam is available at the high-pressure-cylinder exhaust at a pressure high enough to correspond to a temperature of from 300 to 400 deg. fahr. in the average case, while at the same time it is at, or even below, saturation, according to the load.

Follows the literally burning question as to coal-burning equip-

[1] Mechanical Engineer, Sargent & Lundy, Inc. Mem. A.S.M.E.

ment: Shall we use chain-grate stokers, underfeed stokers, or pulverized fuel? The author's views on this subject were expressed in the paper which was read before the Midwest Power Conference in Chicago in February of this year, and it is therefore unnecessary to repeat here the arguments which were put forward at that time.

As to the decision between the chain-grate and underfeed stoker, it has been found desirable to use the latter when a low-ash coking coal is to be used as at the West End station in Cincinnati, and at the Pineville plant of the Kentucky Utilities Company. In

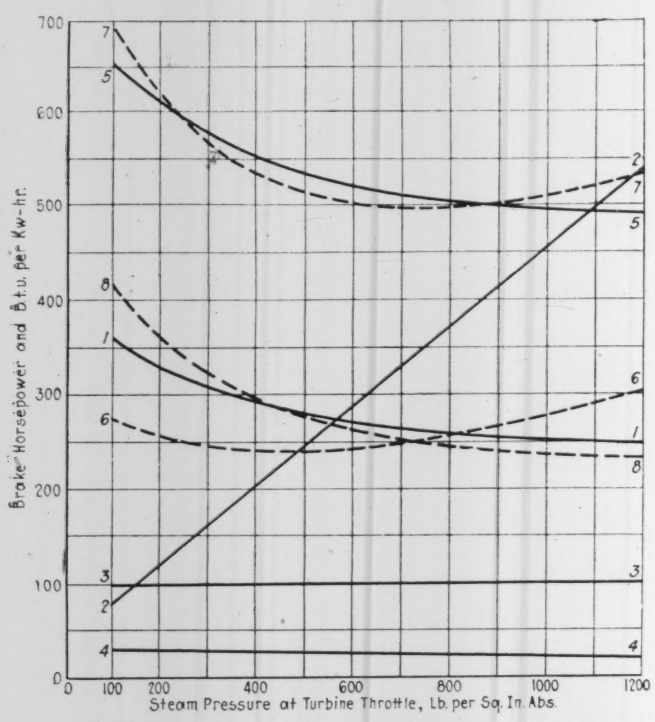

FIG. 2 POWER REQUIRED BY MOTOR-DRIVEN AUXILIARIES OF 30,000-KW. TURBINE UNIT WITH DIFFERENT STEAM PRESSURES AND CONSTANT STEAM TEMPERATURE OF 700 DEG. FAHR. AT THE THROTTLE

Turbine is operated in each case with steam extraction sufficient to heat condensate from 80 deg. fahr. to 210 deg. fahr. and with interstage reheat at and above 600 lb. per sq. in. throttle pressure—all as in curves C-C and D-D of Fig. 1.

1-1: B.hp. required by circulating pump.
2-2: B.hp. required by boiler-feed pump.
3-3: B.hp. required by air pump.
4-4: B.hp. required by condensate pump.
5-5: B.hp. required by forced- and induced-draft fans.
6-6: B.t.u. required by condensing equipment and boiler-feed pump per kw-hr. generated.
7-7: B.t.u. required by all of above auxiliaries per kw-hr. generated.
8-8: B.t.u. required by forced- and induced-draft fans per kw-hr. generated.

the case of the high-ash, free-burning bituminous coals of Illinois, the forced-draft chain-grate stoker is being used, as in the stations of the Commonwealth Edison Company, The Public Service Company of Northern Illinois, and others.

At the Columbia Power station, just outside of Cincinnati, the storage system of burning coal in pulverized form is in use, as it is also in connection with one experimental boiler at the Calumet Station in Chicago.

The unit system of pulverized-coal burning has been installed at Sheboygan, Wisconsin, and is contemplated for some other plants in Texas and Oklahoma, where in all probability it will handle either lignite or the sub-bituminous coal from the north central district of Texas.

In boiler installations where continuously high evaporative rates are expected, with one notable exception, the furnaces are being water cooled either partly or wholly, according to the furnace design, the nature of the fuel, and the form in which it is to be burned; and inasmuch as a great deal remains to be learned in the application of the water-cooling surface to the furnace, these installations are being made comparative as far as possible.

The exception above referred to is that of the Columbia Station of the Columbia Power Company, near Cincinnati, Ohio. This plant, which is burning West Virginia coal in pulverized form, is equipped with air-cooled furnaces. The boilers are operated at an average output corresponding to between 250 and 265 per cent of rating, and during the two years which have elapsed since the plant was started, the furnace maintenance has been negligible.

It must be borne in mind, however, that the problems involved when burning Logan County coal are entirely different from those encountered when Illinois coal is to be used.

Thus at Crawford Avenue Station there are refractory furnaces, furnaces with bridge-wall and side-wall water boxes, furnaces with water-cooled side walls, and others which are entirely water cooled.

There are bare tube surfaces with tubes on various centers, and Bailey walls with refractory-covered blocks and with bare-iron blocks.

This use of radiant heat surface in the furnace, however, emphasizes a factor which the use of high pressures had already brought to the front, which is the evaporator for boiler-feed make-up. The multiple-effect low-heat-level evaporator has been in use for a number of years at Tulsa, Oklahoma, Philo, Twin Branch, and elsewhere, and has long since justified its installation when either bad water, high steam pressure, or a combination of the two, is concerned. Recently, however, the tendency has been to install the evaporator in unit form, rather than as a central water-treating plant.

In other words, a single- or double-effect evaporator is being made an integral part of the regenerative feedwater system of each main unit, to the needs of which it lends itself admirably. It is very flexible as to heat level and pressure range, it utilizes extracted steam as a heating agent, and the applied heat may all be recovered except for the heat in the blowdown and that lost by radiation. In addition, it is a low-priced piece of apparatus, re-

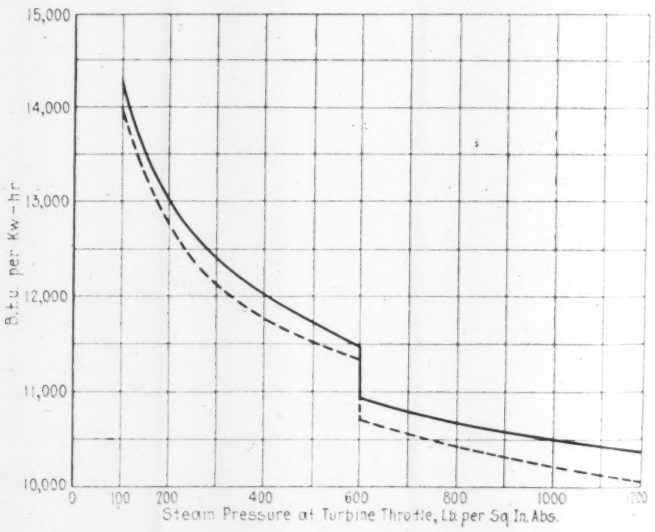

FIG. 3 PERFORMANCE OF 30,000-KW. TURBINE UNIT WITH DIFFERENT STEAM PRESSURES AND CONSTANT STEAM TEMPERATURE OF 700 DEG. FAHR. AT THE THROTTLE

Broken line: B.t.u. per kw-hr. generated, with steam extraction sufficient to heat condensate from 80 deg. fahr. to 210 deg. fahr., and with interstage reheat at and above 600 lb. per sq. in. throttle pressure.
Solid line: B.t.u. per kw-hr. sent out under above conditions.

quiring no additional pumps except for recirculation, and no special condenser as the vapor from the last effect is used to supplement extracted steam in one of the low-pressure-stage heaters. This heater requires no additional surface, as the evaporator should be run off-peak as far as practicable, under which condition extraction pressure and flow are below normal.

Before leaving the feedwater system, mention should be made of the generator air cooler and of the principle of deaerating boiler feedwater. The closed system of generator air cooling, with external fans and either single-stage or two-stage water cooling, has become standard at this time. In the single-stage air cooler the water is taken either from service wells or from the main-condenser circulating-water system. In the two-stage cooler there are two sections of water-cooling surface, the higher-temperature section taking condensate from the main-unit hotwell pumps, and the low-temperature section using either ground water or water from the main-condenser circulating system. This latter system is more expensive and involves more piping than the former, while

the heat recovered by the turbine condensate reduces the low-pressure extraction by a corresponding amount, and therefore does not present any improvement in economy.

With regard to deaeration, there is considerable difference of opinion on the part of operating engineers as to the relative merits of the closed feedwater system and of the flash system of deaeration. The former requires closer attention to the physical condition of the plant than does the latter. To insure the exclusion of soluble gases from the closed system, all main condensers must be free from circulating-water leakage through split tubes or leaking ferrule packings; all closed heaters and connections which operate at pressures below atmospheric must be tight; hotwell pumps must be of the double-impeller central-inlet type having the glands under internal pressure instead of vacuum; emergency cold feed connections to the suction of the boiler-feed pumps should be double-valved, etc.

The flash system of deaeration, in which the separator is the last step in the feedwater system before the boiler-feed pump, allows more margin in the physical condition of the plant. It is not intended to suggest that neglect is allowable, but to point out that the system can continue to supply air-free water to the boiler plant even should a joint fail in a low-pressure-heater steam connection, or a condenser tube split at a time when it is not practicable to shut down the unit.

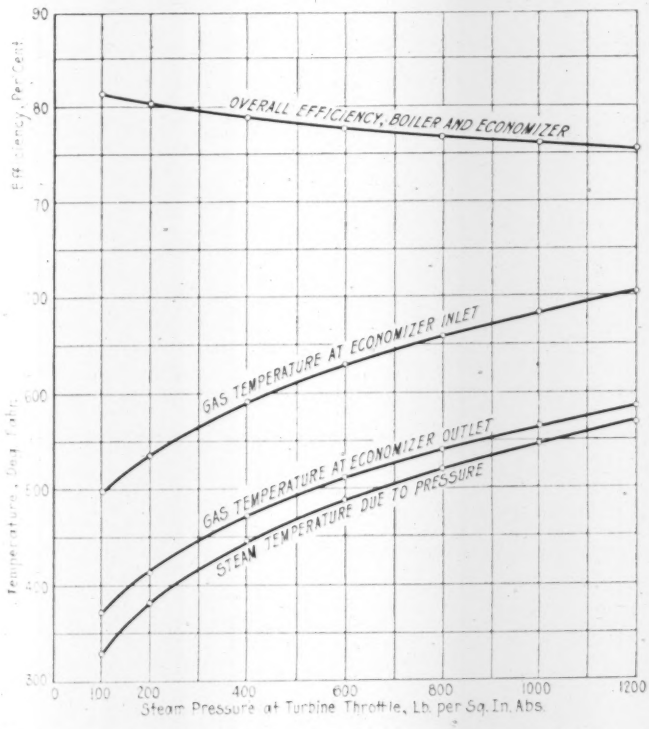
OVERALL EFFICIENCY, BOILER AND ECONOMIZER
GAS TEMPERATURE AT ECONOMIZER INLET
GAS TEMPERATURE AT ECONOMIZER OUTLET
STEAM TEMPERATURE DUE TO PRESSURE
Steam Pressure at Turbine Throttle, Lb. per Sq. In. Abs.

FIG. 4 CALCULATED PERFORMANCE OF "CALUMET" TYPE BOILER HAVING ECONOMIZER SURFACE SUFFICIENT TO RAISE FEEDWATER TEMPERATURE FROM WITHIN 60 DEG. FAHR. OF THE TEMPERATURE DUE TO PRESSURE, TO THE STEAMING POINT

On the other hand, the flash system requires a definite temperature relation between the water outlet of the vapor condenser and the discharge of the final heater to the separator. A variation of five degrees in a terminal difference of twenty-five is allowable under normal load conditions, but not more; and it must be recognized that with the flash system fairly close attention must be given to the temperature relations maintained throughout the deaerator system. This is particularly true on fluctuating loads.

Both of these systems are in use at the Crawford Avenue Station in Chicago, and the results indicate that they are equally satisfactory when given the necessary attention.

With regard to coal and ash handling, each plant presents its individual problems, which are almost entirely the result of purely local conditions such as the source of fuel—whether it is the intention to buy at prevailing prices and so limit the amount of storage to a minimum, or to provide storage space enough so that the operating company may avail itself of favorable market conditions. Again, whether the nature of the coal will allow of storing to a considerable height, whether it must be laid down in shallow piles and vented, whether it must be kept moving or even, perhaps, placed in submerged storage.

In closing this very brief summary of the principal features of the modern power station, a reference should be made to one or two of the more specific points mentioned in Mr. Alden's paper.[2] In enumerating the effects of feedwater heating by steam extraction, for example, Mr. Alden states that with water heated in this way to between 300 and 375 deg. fahr., it is not possible by passing it through an economizer to pick up any appreciable amount of additional heat nor to effect any appreciable reduction in flue-gas temperatures.

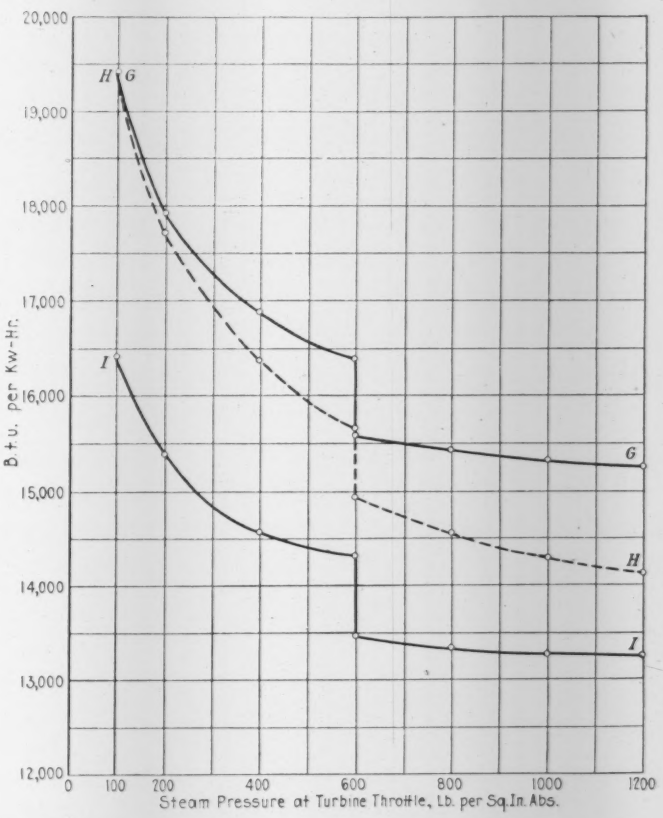

FIG. 5 B.T.U. PER KW-HR. TO TURBINE UNITS WITH DIFFERENT STEAM PRESSURES AND CONSTANT STEAM TEMPERATURE OF 700 DEG. FAHR. AT THE THROTTLE

Turbine is operated in each case with steam extraction sufficient to heat condensate from 80 deg. fahr. to within 60 deg. fahr. of the temperature due to pressure and interstage reheat at and above 600 lb. per sq. in.

Steam is supplied by "Calumet" type boilers having economizer surface sufficient to raise feed temperature 60 deg. fahr. to steaming point.

I-I: Station B.t.u. per kw-hr. generated with boiler and economizer efficiencies as in Fig. 4.

G-G: Station B.t.u. per kw-hr. sent out with operating efficiency of 90 per cent.

H-H: As G-G, but with air-heater surface sufficient to maintain stack temperature of 415 deg. fahr. in each case.

This is true of low-pressure plants, but in the 600-lb. plant, where the saturated-steam temperature due to the pressure is 500 deg. fahr., it is quite practicable and economical to raise the temperature of the feed from 375 to 450 deg. fahr. in the steel-tube economizer. This 75-deg. rise would be equivalent to a decrease in outlet-gas temperature of about 187 deg. fahr., and would involve a quite reasonably small amount of economizer surface. If, furthermore, an economizer of the steaming type is used, as at Crawford Avenue, the water may be raised to the boiling point of 500 deg. fahr. in the economizer.

To the advantage of feedwater heating by steam extraction from the main unit, as listed by Mr. Alden, may be added the fact that with an ultimate feedwater temperature of 350 deg. fahr. the amount of steam passing entirely through the turbine to the main conden-

[2] Present Tendencies of Steam-Station Design, by V. E. Alden, published in the June issue of MECHANICAL ENGINEERING, p. 603, and presented at the same session of the Kansas City Meeting as the present symposium.

ser is about 77 per cent of the flow to the throttle, and inasmuch as the additional steam flow to the throttle for a given output is only about 60 per cent of the weight of steam extracted, the condenser may be materially smaller for the turbine operating on the regenerative cycle than would be required for the same output

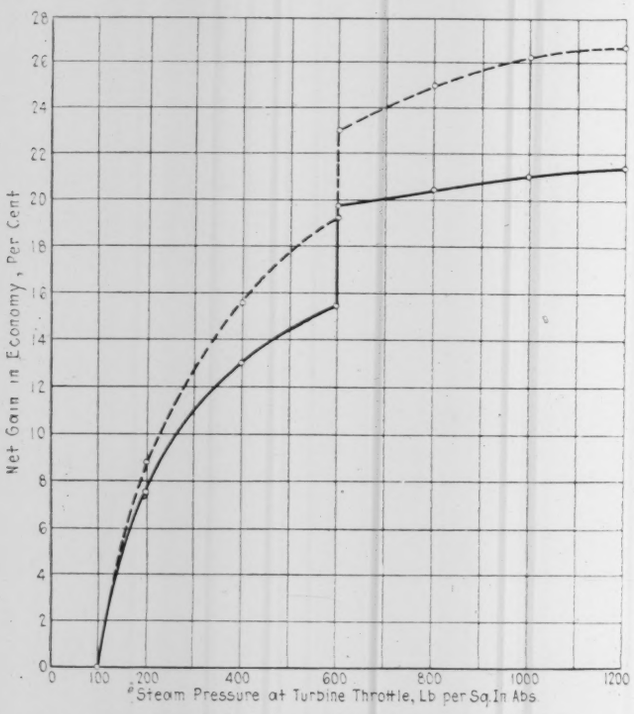

FIG. 6 NET GAIN IN ECONOMY AT DIFFERENT STEAM PRESSURES AND CONSTANT TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR., EXPRESSED AS PERCENTAGE OVER RESULTS OBTAINED AT 100 LB. PER SQ. IN. ABSOLUTE AND 700 DEG. FAHR.

Solid line corresponds to G-G, of Fig. 5.
Broked line corresponds to H-H, Fig. 5.

if all of the steam were expanded down through the turbine to condenser pressure.

Also the amount of circulating water required is reduced in the same ratio for a given absolute condenser pressure, so that with the regenerative cycle more kilowatt capacity may be installed in a plant where the supply of circulating water is limited, than in the case of the non-bleeding type of turbine.

Operation Problems Encountered in Boiler Operation at Omaha

BY CHARLES F. TURNER,[3] OMAHA, NEB.

IN THE Omaha station of the Nebraska Power Company, there are three 1500-hp. B. & W. cross-drum boilers and one 2000-hp. B. & W. cross-drum boiler. Each boiler is equipped with a Coxe chain-grate stoker, a Foster side-wall radiant superheater, and a Duratex counterflow economizer.

Fuel used in this plant is bituminous slack from Illinois, west Kentucky, and Kansas. Two characteristics of certain grades of Kansas coal are low ash content and clinkering quality of the ash. The clinkers form on the grate in a molten mass and cling with such tenacity that it became necessary to construct a scraper at the ashpit. This scraper is made of slabs of cast iron about 12 in. by 18 in. by $1/2$ in., each slab being hinged on a common support rod. The supporting rod is in turn fastened beneath the return shaft at the edge of the ashpit, and the scrapers drag the clinkers from the grate into the ash hopper.

STOKER DRIVES

Each boiler is served by two grates and consequently has two stoker drives. Stokers are driven by direct-current motors, and one of the first experiences showed the necessity of an auxiliary drive to serve in an emergency. For this purpose two alternating-current motors are mounted on low trucks, to facilitate moving, and one of these motors can be wheeled into place, bolted to the floor, and put in operation in less than two minutes.

ECONOMIZERS

Our first difficulty with economizers came after six months' service when we found internal pitting progressing at an alarming rate. Increasing the feedwater temperature from 180 to 210 deg. fahr. decreased the corrosion to a minimum by driving off the dissolved oxygen.

Cleaning the outside surface of the tubes presented the next problem. The steam pressure on the soot blowers at the time was 200 lb. and did not thoroughly clean the tubes. Washing with water at intervals of two weeks cleaned the surface down to the bare steel. When washing an economizer, the boiler must be operating at high rating and the fans stopped. After washing thoroughly the boiler is brought to high rating again and thus the economizer is quickly dried out.

During 1925 and 1926 the steam pressure was raised from 200 to 325 lb., and it was discovered that the economizers did not get dirty as heretofore due to the higher pressure on the soot blowers.

Some trouble was experienced on one economizer when a tube roll started a small leak. This leak could not be detected from the outside, and as a result a nest of 13 tubes corroded so badly in this spot as to cause their removal. We believe more adequate provision should be made for inspecting the ends of tubes next to the header boxes, and we are considering placing doors to accomplish this, up and down the ends of the economizer.

In January of this year a general clean-up and overhaul of the economizers disclosed excessive pitting around handhole caps and boxes, due to seepage of water through gaskets, with the resulting formation of acid when soot and water come in contact. Headers and caps have now been painted with graphite and boiled linseed oil to prevent this corrosion.

SLAGGING

In operating these boilers at from 200 to 300 per cent of rating we found it impossible to maintain mechanical soot blowers in the lower part of the first pass. At the same time this is a vital part of the tube surface, and with high ratings the slag forms on the lower or drop tubes quite rapidly. Therefore a steam hand lance was used to supplement the work of the mechanical blowers. But a steam hose, whether metal or rubber, is a hot, dangerous tool at best, and with the pressure at 300 lb. the danger is increased proportionately. We decided to try water, and after some experimenting found that city water at 100 lb. pressure when forced through a nozzle having a diameter of $5/32$ in. would do the work almost as fast and certainly cleaner than steam at 300 lb. Also with a smooth, straight nozzle three inches long it is possible to clean slag from a tube 15 to 20 ft. from the end of the 4-ft. lance.

The side-wall radiant superheaters were equipped with special soot blowers, but these soon burned off and we have been very successful in using a lance made of $1/2$-in. pipe having a nozzle at the end drawn down to about $3/16$ in. diameter. In this lance we use compressed air at 100 lb. pressure, and by entering it through a narrow door in the back wall close to the superheater surface we are able to blow all slag from the exposed surface in a very few minutes.

SIZING AND TEMPERING COAL

There are two more items that may be of interest to some operators in regard to preparation of coal for burning. First, all coal is tempered before being run through the crusher, and is then crushed to such size as to pass through a $1/2$-in. screen. In the tempering it is our experience that when the crushed coal can be molded in the hand without making the hand wet, the moisture content is just right to give an even air distribution and thus obtain best results in the furnace. It may be interesting to some to know that after repeated tests we have found that 14 per cent total moisture gives the desired condition. In other words, if a coal has 8 per cent moisture we must add 6 per cent in order to get the right consistency, but if it has 12 per cent moisture, we need add only 2 per cent.

[3] Chief Operating Engineer, Nebraska Power Company.

The Use of Pulverized Fuel at the Big Sioux Station of the Sioux City Gas and Electric Company

By K. M. IRWIN,[4] PHILADELPHIA, PA.

THE Big Sioux Station of the Sioux City Gas and Electric Company, which was put into operation in June, 1925, was designed and laid out for the use of powdered fuel, which was thought to be particularly applicable in this general locality due to the wide variation in types of coal which could be purchased. The engineers felt that any other method of firing would not be as flexible or as adaptable to the changing qualities of coal which could be obtained in this market.

The bin and feeder system is employed, and the plant is equipped with Raymond-type mills, a Fuller-Kinyon conveying system, and Lopulco burning feeders and burners. The boilers are 1100-hp. Stirling boilers and the furnaces are designed with hearth screen and air-cooled side walls. The additional boiler now being installed will, however, have water-cooled side walls of the Murray type. This plant has successfully burned coal from Iowa, Illinois, Indiana, Kentucky, Arkansas, Oklahoma, and Colorado. The only serious difficulties experienced have been with the Oklahoma coal, which, strangely enough, has the lowest moisture content and the highest B.t.u. of any of the other coals used; the one with the lowest B.t.u., of course, being the Iowa coal, which has a heating value of 8700 B.t.u., with a moisture content as high as 16 per cent and with ash as high as 20 per cent. By drying this coal to approximately 12 per cent we have experienced no difficulties in the mills, transportation system, or the feeding and burning equipment, but with the Oklahoma coal, which has a heating value of approximately 13,500 B.t.u. and moisture in the neighborhood of 4 per cent, trouble has been experienced with clogging in the feeders, uneven feeding, and, at times, packing of the coal so solidly as to stall the feeder motors. We are now attempting to correct this by bypassing the coal through the driers, introducing flue gas directly into the mills, and also by venting the pulverized storage bins by taking a lead to the inlet of the primary air fans and in that way attempting to hold the air in the storage bins above the saturation point.

When the plant was first started, we experienced some difficulty with clogging in the transportation system. Much of this difficulty has been overcome, especially in the colder weather, by introducing a cooler, a separator, and then a heater in the compressed-air line going to the Fuller-Kinyon pumps.

Difficulty was also experienced in the early operation with the dust coming from the mills' cyclone vents, and the design was changed to wet this dust with steam and then wash it out with water, having the final vent connected to the stacks. Since that time the dust nuisance has been materially reduced.

Considerable washing has been experienced on the front walls of these furnaces, and now, at the end of nearly two years, it will be necessary to make some extensive repairs to these walls. On one of the boilers this is being attempted by the use of the cement gun. This proves to be a very low-cost method of repairing the walls, but the boiler since repair has not been in operation a sufficient length of time to make a definite statement as to the durability.

Trouble was also experienced at one time with the flames sweeping to one side of the furnace, causing side-wall washing. This has been overcome by a more careful adjustment of the spring loading on the Bailey side-wall dampers, and by the installation of deflectors for the secondary air in the air passage at the side openings and by deflecting more air to the side walls. Deflectors of the Combustion Engineering Company's design have been installed in the burners of one of the boilers, and the furnace conditions of this boiler have been appreciably improved; the ignition is faster, the furnace is clearer, and the ability to keep the flames down to the water screen at higher ratings is increased. We were disappointed, however, that these deflectors did not decrease the front-wall washing.

It is interesting to note that during the week of March 14, 1927, the plant utilized approximately 250 tons of west Kentucky coal, 250 tons of Illinois coal, 450 tons of Iowa coal, 80 tons of Arkansas coal, and 50 tons of Oklahoma coal.

The weighted average B.t.u. per pound of coal burned was 9970, and the average B.t.u. per net kilowatt-hour generated was 17,427. The average boiler efficiency over the week was 84.6 per cent. We consider these results extremely good, considering the diversity of coal burned and the type of load.

Pulverized Coal in the Middle West

By E. H. TENNEY,[5] ST. LOUIS, MO.

AS A BRIEF contribution to this discussion of steam-station development in the Middle West, I would like to touch upon the subject of pulverized coal because it seems to be of especial interest to most of us at this time. The differences between the eastern and midwestern coals seem to be more pronounced when burned in powdered form. The high moisture content of the midwestern coal involves drying, milling, and transporting complications, while a high volatile content involves special treatment in drying and in furnace design. The high percentage of ash which these coals contain also acts as an abrasive in mills and in transport lines, while its low softening temperature, due to high iron and lime content, acts as a flux which is particularly destructive to refractories and is the cause of slag.

In the evolution of combustion-equipment design in our St. Louis plants we have endeavored to meet these conditions to increasingly better advantage.

At the Cahokia Station more effective drying, milling, and transporting equipment has solved most of the problems of preparation, although the maintenance costs on this equipment still comprise one-fourth of the station total. Erosion of side-wall refractories has been eliminated only by the use of water-cooled walls. The fin-tube type is satisfactory except that it is difficult to avoid carbon loss and smoke at reduced ratings. A wall of water tubes recessed in brick is proving to be the most satisfactory construction. Combustion conditions with vertical, straight-shot burners are not satisfactory with this grade of coal due to retarded combustion and consequent forcing of the hot zone into the ashpit. Our experience has demonstrated that there is a great advantage in supplying larger percentages of combustion air at or in the burner, and giving the air and fuel mixture sufficient turbulence to cause early and complete ignition. This results in lifting the hot zone away from the ashpit and eliminating incomplete combustion in the furnace and the formation of slag.

Smoke-abatement work has been particularly active in St. Louis for several years, and as a matter of public example we have been going to a good deal of trouble and expense to burn this Illinois coal smokelessly.

Experiences with Unit Mills

About a year ago twenty boilers at Ashley Street Station were equipped with unit mills and water-cooled furnaces and are giving very wonderful results. In this installation a limited space available for the furnace necessitated short flame lengths and a high rate of flame propagation. Horizontal burners admitting more air with the fuel and imparting turbulence to it, proved an effective way of accomplishing this. As a result of these changes the steaming capacity of each boiler has been increased 50 per cent and the combined boiler and economizer efficiency brought up from a previous 70 per cent to the present 82 per cent. This installation operates smokelessly and without slag in the ashpits and with equal dependability to that of the boilers in Cahokia Station which are equipped with the storage system.

An experimental installation of unit mills was made last summer on one of the Cahokia boilers on which tests had previously been made with the storage system. The installation consisted of two 8000-lb. mills with four horizontal burners of the turbulent type. On test it was demonstrated that substantially the same economies could be obtained as with the storage system, and with the

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horizontal burners the ashpit was shielded by air and coal from full furnace temperature, resulting in the elimination of overheating and slag.

A careful study of the comparative first costs and operating economies of the unit versus the storage system has resulted in the adoption of the unit system for three 1801-hp. boilers now about to be erected in the third section of Cahokia. There are to be two simplex mills per boiler, each with a capacity of 15,000-lb. per hr. These will deliver coal through four horizontal, forced-draft, turbulent burners supplied with preheated air. The furnace side walls are to be of the recessed-water-tube and refractory type of construction and there will be no water screen. There will be iron-shrouded water-wall construction for the rear wall, ashpit, and front-wall arch above the burners.

Based on experience with both unit and storage systems in St. Louis, it can be confidently stated that the two may both be successfully used with midwestern coals, but that there are certain advantages in favor of the unit system which may be summarized as follows:

1 Lower investment costs, due to elimination of storage bins, driers, separators, and transport system
2 Equal furnace efficiencies with less operating and maintenance labor
3 Greater simplicity of installation
4 Cleaner, quieter, and probably safer operation.

Improvements and Developments at Northeast Station of the Kansas City Power and Light Company

BY J. A. KEETH,[6] KANSAS CITY, MO.

MR. ALDEN'S paper on Present Tendencies of Steam Station Design outlines generally many of the developments that have been made at the Northeast Station of the Kansas City Power & Light Company. These developments have been made with at least one, and in many cases more, of the ultimate objects in view which he has outlined. Certain problems are peculiar to individual stations, due perhaps to geographical location, kind of fuel, source of circulating water, and characteristics of the system of which the station is a part. A brief statement of changes that have been made in this station to meet these peculiar conditions as well as a summary of changes along the lines mentioned by Mr. Alden, may be of interest.

The Northeast Station was designed by Sargent & Lundy and represented, we believe, at the time it first went into service in 1919, the best available without making any more or less radical departures from the conservative state of development at that time. It was a station of moderate pressure and temperature, 250 lb. and 250 deg. superheat, with units of a size that were at that time considered reasonably large. The plant was a chain-grate-stoker station designed primarily for the use of low-grade midwestern bituminous coals. Induced draft was provided so that no difficulty was experienced in obtaining efficient boiler operation at 250 per cent of rating. Boilers were provided with the double system of economizers mentioned in Mr. Alden's paper. The original three turbines were not equipped with extraction. Most of the auxiliaries in the station were motor-driven.

This last feature along with the fact that the Northeast Station was the only plant operating on the system, soon made it evident that some means must be provided for starting or maintaining auxiliaries in case of complete shutdown resulting from any of the many possible causes, chief of which was lightning. Accordingly in an addition made to the plant in 1922, a 1250-kw. non-condensing house turbine was installed. This equipment was installed in such a manner that instantaneous and automatic protection was provided for certain necessary auxiliaries normally without any effect on the heat balance of the station. Under normal conditions the generator end of this unit is operated as a synchronous motor tied to the station auxiliary system through an overload and low-voltage relay. The throttle valve of the turbine is always open, but the setting of the governor for a speed slightly below system frequency keeps the admission valves closed until some interruption occurs to the system. The interruption causes one or the other of the relays to operate, cutting the unit free from the auxiliary transformer system. The frequency of the connected auxiliary system drops to the point where the turbine admission valve opens up and picks up the connected load without interruption. A small supply of saturated steam keeps the turbine wheels cool during motoring operation.

Included in and since the 1922 addition to the station, two 30,000-kw. turbines have been added, as well as four additional boilers. Unlike the first three units, these later machines were provided with extraction at a point near atmospheric pressure. Sufficient steam was bled from these turbines to heat their condensate to approximately 165 deg. fahr. The installation of electric boiler-feed pumps and the corresponding reduction in auxiliary exhaust steam pointed the way to still further extraction. One of the original 20,000-kw. turbines was remodeled last year to provide for extraction from the nineteenth stage of sufficient steam to give a condensate temperature approaching that of the 30,000-kw. units. It is planned to make similar changes in the other two 20,000-kw. units this year.

It is believed that this station was one of the first to do away with the unloading valves for releasing the leak-off from the high-pressure packing of the turbines into one of the lower stages. This not only eliminated the undesirable effect of admitting high-temperature steam to a cool part of the turbine, but recovered all of the heat, a large part of which was previously lost. A rise of from 10 to 20 deg. in the condensate temperature was obtained by the use of gland-steam heaters.

Along the lines of reducing auxiliary power use in the station, the original hydraulic vacuum pumps are being replaced with steam ejectors. These not only reduced the auxiliary power, but produced better vacuum when the results of the old equipment were limited by the temperature of the hurling water. Cross-connections have been provided between circulating-water systems to permit the use of less water during the periods of the year when the river temperature is low.

The use of vertical condensers, as installed in this plant, presented a problem in obtaining a condensate temperature reasonably close to the steam temperature. It seemed impossible to design this type of equipment in such a way as to prevent a certain amount of refrigeration effect. This problem was solved on the last unit installed by the use of an external condensate heater. Steam from the exhaust connection to the condenser is induced into an open-type heater operating under high vacuum, through which the condensate is pumped. The condensate is thus maintained within 1 to 2 deg. of the steam temperature where formerly a drop of 6 to 8 deg. existed.

A surface-condensing plant using Missouri River water has problems of a nature probably not encountered on any other river. The water not only contains large percentages of sand in suspension, but also quantities of vegetable matter and small pieces of water-logged wood almost impossible completely to eliminate by revolving screens in their present state of development. In order to eliminate most of the unit outages for condenser cleaning, flush gates or butterfly valves were designed and built into the partition between the entering and leaving condenser water boxes. The grass and water-logged wood which obstructed the entrance to the tubes in the first pass of the condenser could thus be quite effectively flushed directly out the discharge through these gates without any interruption to the service. The use of a venturi type of ferrule flush with the tube sheet also greatly improved conditions by restricting the tube entrance to the point where any material which could pass through the ferrule could pass through the tube without any danger of wedging. The flush surface gave no projections for debris to adhere to, and made the flushing operation much more effective. In the last condenser installed, arrangements are provided for complete reversal of flow without shutting down the unit. All of these improvements to the condenser permitted operation with better vacuum and resulted in less shutting down and starting of equipment.

Experience with intakes and intake screens in Missouri River

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service has dictated changes and improvements which would require a complete paper for treatment.

Progress and development at this station have not been confined to the turbine equipment. All boilers with the exception of the first six have been equipped with forced-draft chain grates which have permitted better efficiencies at higher ratings. An experimental installation of pulverized coal is now being made under one boiler, replacing some of the old induced-draft grates. The use of certain varieties of midwestern coals for pulverized firing is new, and accurate information as to the results possible can be obtained in no other way than by actual trial. No prediction as to results is being made. The experimental unit is being equipped in a most modern method. Exceptionally large furnace is provided. Walls are made up of fin tubes, radiant superheater, and hollow-tile walls. The fuel will be prepared by two 5-ton unit-type impact mills.

Three boiler units have been equipped with air preheaters, and a fourth will soon be similarly equipped. The first air preheater ordered for a central station in this country was ordered by this company in 1922. The results proved so gratifying in the burning of low-grade fuels that its use is sure to become general for this class of service.

The use of air heaters forced the issue of radiant-heat superheaters. The efficient operation of the furnace with heated air so reduced the volume of flue gas that additional superheater surface was required to maintain desired steam temperature. The only space available for this additional surface required was in the furnace side walls. The radiant superheaters have proved so successful that plans are now under way to raise the steam temperature of the entire station to approximately 700 deg. by the installation of additional radiant superheaters in all boilers. This will result in a material improvement in the overall economy of the plant.

In the final effort to keep this station abreast with modern development and economy, plans are now being drawn up for the installation of two 1400-lb. boilers and one 10,000-kw., 1200-lb. turbo-generator. This company gave serious consideration to this idea in 1923 before any similar installations had been made. However, the load conditions were not such then as to permit the installation. The growth of the load has now brought the minimum system demand to a point where this high-pressure unit and its necessary complement of low-pressure units can be operated at a high load factor, practically a base load. It is firmly believed that the addition of such high-pressure equipment will become quite general for improving the economy of some of the better class of older stations.

Many other changes and improvements have been made, some resulting in reduction of labor and maintenance, others directly effecting economy, while still others have been made to improve reliability or reduce ultimate investment costs. Most of these except those mentioned are entirely too peculiar to local conditions to justify mentioning, while perhaps others would be of interest if time permitted their discussion. Few improvements originate in any one engineering office or individual company, but are largely the result of widely separated experiences and needs, to which every operating company and designing engineer should, and usually does, contribute.

Steam-Station Development in Middle West

By C. S. TOMLINSON,[7] PARSONS, KAN.

MR. ALDEN'S paper on the tendencies of steam-station design has given a clear picture of the development of stations throughout the country. This same process of development has gone on in the Middle West, but with conditions peculiar to this section. There have been no very recent developments in this section in the way of unusual efficiencies or unusual temperatures and pressures. The large industrial centers in the East are practically all located on streams or bodies of water which will furnish plenty of condensing water for the generating stations, and the stations are accordingly for the most part located close to the load. In this section of the country there are no lakes and very few rivers which do not entirely dry up at times. This of course restricts the location of stations to these streams, and even then it is often necessary to supplement the natural water supply with some sort of cooling system.

The load to be served by central stations in this section is scattered in small blocks over a wide area. For instance, in Kansas flour mills and grain elevators are located in almost every town; the oil fields, refineries, and pipe-line pumping stations are at various points over most of the eastern half of the state, and so on with the cement plants, quarries, brick and tile plants, coal mines, and lead and zinc mines. The fuels available are somewhat more varied than in almost any other section of the country. To the south, particularly in Oklahoma, there is natural gas which, close to the fields and even at some distances, can be purchased for industrial fuel at very reasonable prices. The petroleum industry provides a large amount of fuel oil, and the coal fields are numerous and near at hand.

The peculiarities of steam-station development in the Middle West are scarcity of desirable plant sites due to lack of condensing water, scattered load, and variety of fuels.

Scarcity of Condensing Water

On the Missouri River the stations have sufficient water but must contend with the large amount of sand and silt which the river carries. This has a habit of lodging in condensers, in intakes, or sometimes even forms a bar which completely isolates the intake from the channel. On account of this trouble the Kansas City station has used vertical condensers equipped with sand pumps to clean the lower water boxes. The intakes, tunnels, etc. are kept clear by the use of sand pumps or other dredging equipment. One compensation to the operating man under these conditions is that there is usually no condenser-tube cleaning to be done. The sand in suspension in the water keeps the tubes well polished.

Where the supply of water is not sufficient, either a spray pond or a large pond using surface cooling is installed. A spray pond may be installed in a moderate space, whereas a surface cooling pond must be much larger, and I believe the spray pond has been more generally adopted. The water is pumped to the spray nozzles under a pressure to give 7 lb. at the nozzles. With this pressure the water is quite finely atomized and the cooling is effected both by direct contact with the air and by evaporation. It is customary to place the nozzles in groups, and the groups in lines spaced far enough apart to make air lanes. This matter of air circulation is particularly important, for no matter how well the water may be sprayed, satisfactory cooling cannot be obtained unless the water and fresh air are thoroughly mixed. At one installation with which I am familiar there was one portion of the pond equipped with an old-style nozzle spaced along one pipe line seven or eight hundred feet long and another portion of the pond with newer nozzles mounted at regular distances along laterals from the main line. On individual test the new nozzles gave two or three degrees more cooling than the old style, but taking the two sections of pond as a whole, the older gave better results unless there was a stiff wind blowing into the air lanes of the new part. Mist blowing off from a spray pond may be very objectionable to the surrounding property. Louver fences will stop much of this mist, but correspondingly shut off the circulation of air. With high winds the moisture will drift through and over the fences and travel several blocks. Another difficulty experienced with spray ponds is that during the season of the year when very little make-up water is added, the concentration of solids in the water becomes greater and greater, and these solids are precipitated out as scale in the condenser tubes. This scale is sometimes very hard and tough, resulting in a difficult problem to keep condensers in shape. Finally, there is the cost of pumping. At our Wichita station the power used for spray pumps during 1926 amounted to 1.57 per cent of the station output. There was not a month that the natural flow of the stream was sufficient.

As stated before, a surface cooling pond requires a large amount of room, but where it can be employed it is not subject to some of the objections of a spray pond. There is no mist to blow, the volume of water is so much greater that serious concentration is not as likely, and the pumping cost is almost negligible. It is

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my understanding that the Valmont station in Colorado depends almost entirely upon this method of cooling. The Riverton, I believe, and Neosho stations in southeastern Kansas use this system to a certain extent. In spite of the objections to cooling systems they are being used more and more, and are made to work where the demand justifies the expense.

Scattered Load

The scattered load which was the second peculiarity of the Middle West is in one way a help. One station may serve a good many square miles of territory, and the best plant site in that territory can be developed without much if any increase of transmission loss.

Variety of Fuels

Of the fuels available in the Middle West, gas and oil are much the most satisfactory from the standpoint of convenience. Gas is perhaps the nicest of all, as it is only necessary thoroughly to mix it with the air in the furnace to get complete combustion and have very little excess air. There are several burners which do this, but in many places it is necessary to have some other fuel in case the gas fails. The combination of gas and oil burning has proved a very satisfactory method of taking care of the situation.

There are two general types of oil burners—steam and mechanical. In the use of either it is necessary that the oil be heated in order to properly atomize it. The heaters used are similar to a closed feedwater heater and regulation may be by hand, by temperature controller or viscosity controller. In the steam burner the energy of the steam is used to blow the oil out usually in a fan-shaped spray. The air is admitted from underneath, and it is necessary to have a very long flame travel to complete combustion before reaching the boiler tubes. In the mechanical burner the oil is pumped to the burners at from 100 to 200 lb. pressure and delivered from the burner tip in a very finely divided cone-shaped spray. The air is admitted around this and from the shape of the air vanes given a whirling motion, thus intimately mixing the air with the oil. Combustion takes place quicker and so requires smaller furnace volume, and there is no blast action as with the steam burners. These burners give somewhat better efficiency than steam, and are generally used in central stations burning oil. They are also built as a combination oil and gas burner. The oil delivery pipe and sprayer plate are replaced by a gas pipe and nozzle. The change can be made in a minute or two.

Gas gives a low furnace temperature and is therefore very easy on brickwork. Oil gives a high furnace temperature, and the operator must not cut down excess air too far or the brickwork may be melted. Brickwork stands up very well ordinarily because there is no molten ash or slag to flux the walls. Spalling may occur if cold air is allowed to hit the hot walls.

The theoretical efficiency obtainable with gas is better than with oil, but our experience with both has been that there is only a very slight difference, and that in favor of oil.

Coming down now to coal, I shall confine myself to our experience with southeast Kansas, or, as it is commonly known, "Cherokee," coal on Taylor underfeed stokers at the Neosho station. These stokers are type H-12 retort, 29 tuyeres long, and have been in service three years. Two of these stokers have steam-operated dump plates and one has clinker grinder in a 5-ft. pit. The coal used has been almost entirely slack and runs about 11,000 B.t.u. as received. This coal is extremely variable having sometimes a heating value below 9000 B.t.u. and sometimes as high as 13,000. Moisture has varied from 6 to 16 per cent, ash from 10 to 21 per cent, and sulphur from 2.7 to 5.8 per cent. These figures only give an idea of how the coal varies in its behavior. Some burns very slowly, some fast, some cokes, some does not, some clinkers, some does not, and so on. However, with its poor quality and variations we have been able to burn it at ratings to the limit of our uptake draft, with results which we consider good.

During a sixteen-month period in 1925 and 1926 the boilers were operated at an average rating of 190 per cent and maximum of 250 per cent. The boiler efficiency attained was 76 per cent. The combustible in the refuse was 22 per cent and the CO_2 at the uptake was 12.6 per cent. The boilers are 12,000-sq. ft. Stirlings with a 3000-sq. ft. integral economizer.

The first trouble encountered with the operation of the stokers was the formation of heavy clinkers which would adhere to the lower end of the tuyere stacks and to the extension grates. Moving the pushers of the stoker further into the furnace made the retorts shallower and gave the pushers a better chance to keep the fuel bed moving. So long as the fuel bed is kept in motion there is no serious clinker trouble. To keep the fuel bed free from holes and to burn out the refuse before dumping it into the ashpit is the problem. Except with the worst of the coals it is possible to do this by proper adjustment of the stoker. It means that the fireman must be constantly alert to change adjustments with each change of coal.

The clinker pit of the stoker grinder gave the most trouble. The clinker would move off from the end of the stoker in a sheet which would sometimes lodge against the bridge wall and hang there. If the fire was carried low enough in the pit to prevent this, there was danger of getting it too low and burning the stoker. Various water sprays were tried to make the clinker more porous and brittle so that it would break up and go down. A set of sprays in the bottom tuyeres finally accomplished the desired result, but at the expense of increased combustible in the refuse. On the clinker grinder more uniform conditions can be maintained than on the dump-plate stokers, so that brickwork trouble and slag trouble are less. The improvement in these about balances the loss in the refuse. We are still working on the problem and hope to improve the present situation. In a future installation a deeper pit would be used, and it is believed that if water were still necessary it could be introduced further down so as not to quench the refuse as soon as now.

The ash from all this coal has a low melting point, in the neighborhood of 2000 deg. fahr., and when in a molten condition acts as a flux for firebrick. The chief furnace trouble has been the wall directly over the top of the stoker. At present we are trying some cast-iron air boxes at this point. Sufficient air along this wall stops the fluxing action, and with proper fuel-bed conditions some air does not appreciably increase the excess air at the uptake. The bridge walls have had a life of over a year, most of the deterioration occurring at times of poor coal and bad fires. The side walls have never been rebuilt but patched up six to ten feet above the stokers. The use of Carbofrax brick for several courses above the side-wall air backs of the stoker helps materially in getting rid of clinker formations at that point.

Slag formation on the first pass and superheater tubes gives trouble. The trouble becomes much worse with some coals than with others. Under usual conditions a boiler will operate 30 days before it becomes plugged up so badly that the draft loss is serious. After the boiler has cooled down, most of this slag can be knocked off easily. Some, however, sticks very tightly to the tubes and has to be hammered off. This condition is particularly bad in the superheater because of the small tubes and close spacing.

One characteristic of the Cherokee slack is its readiness to pack. In the storage piles or bunkers as coal is removed, that remaining will stand nearly vertical. This also happens in the stoker hoppers—it is one of the difficulties of keeping a good fire. When it does not feed at all it is readily noticed, but when it feeds partially it will cause a ragged fire and is hard to detect.

We have stored this coal in piles six feet deep without serious fires. In storing it is important to mix the coal as well as possible.

Summing up our experience with the underfeed stokers and Kansas coal, we should not want to undertake a rate of combustion much in excess of 45 lb. of coal per square foot of grate area per hour. With the use of a water-cooled bridge wall and air blocks in the front wall we should anticipate comparative freedom from furnace troubles. Improvements in the design of stokers ought to help the slag trouble. Altogether the stokers ought to give a very good account of themselves in comparison with any other system of firing, and we should give them very serious consideration for a future installation.

* * *

V. E. Alden, in closing the session said that the comments which Mr. Tenney and Mr. Collings had made simply served to emphasize the difficulty of generalizing about a problem which was so involved as that of power-station design. Mr. Tenney was per-

fectly correct in stating that for midwest conditions his (Mr. Alden's) estimate of a decrease of 0.055 lb. of coal incident to improved methods of combustion was too low. He knew that based on conditions in St. Louis with low-grade coal, Mr. Tenney had been able to improve his efficiency, not by the relatively small amount which was indicated in the paper but he had been able to raise the boiler room efficiency from 70 per cent to some 82 per cent by improved methods of firing. The paper had been written with the background of eastern experience in mind, using a relatively good grade of coal. Mr. Tenney had also referred to the effect of water walls on efficiency.

Based on eastern conditions, water walls were mainly effective in permitting us to work boilers and furnaces at higher ratings and keep them on the line more hours per year. Conditions in the West with low-grade coal actually permitted operation with less excess air if water walls were substituted for brickwork and thus effected an improvement in boiler efficiency.

Mr. Collings had commented on the further advantages of the use of regenerative cycle, but he was no more enthusiastic about the use of regenerative cycle than was Mr. Alden, and if the latter had allowed himself free rein, he might have talked for nearly an hour on that alone. The use of bleeding for feedwater heating got the moisture out of the lower stages of the turbine—or helped to, at any rate—reduced turbine maintenance, and kept the turbines on the line more hours per year.

Mr. Collings had commented on the use of the high-pressure economizer in connection with the heating of the feedwater to relatively high temperature by means of stage bleeding. It was true, of course, that with steam pressures of 600 or 1200 lb., it was possible to do things which could not be done with steam pressures of only 400 lb. Mr. Alden had in mind the Edgar station in Boston where the feedwater would be heated to 420 deg. by means of bleeding the turbine and a high-pressure economizer would be used to raise it to still higher temperatures.

It was to be borne in mind, however, that it was only with a high load factor, and where fuel cost was high that the complication of using both economizers and air preheaters was permissible.

It might have seemed from the paper and from some of the things that others had said that it was comparatively easy to reduce the fuel consumption from 1.6 to 0.9 lb. per kw-hr. This was not true. It was the result of a great deal of careful work. To get the perspective, it should be borne in mind that there was probably no single station in Europe that was doing better than 20,000 B.t.u. per kw-hr. There were only two in England that were getting around 19,000 B.t.u. per kw-hr.

This should be considered when one was trying to gage the relative value of the job which was being done here where there were many, many stations that were doing better than 17,000 B.t.u., and one station that had generated power for less than 13,000 B.t.u. per kw-hr. for some three or four months.

The Influence of Elasticity on Gear-Tooth Loads

Progress Report No. 5 of the A.S.M.E. Special Research Committee on the Strength of Gear Teeth

THIS progress report gives the second of a series of studies of the influence of elasticity on gear-tooth loads. Most kinematic calculations are based on the reactions of rigid solids. When the velocities of the moving parts are low, this method gives a reasonably close measure of the conditions; but as the velocities increase, the influence of the elasticity of the materials plays a much larger part, so that the assumption of rigid materials does not give a true measure of the actual conditions.

Progress Report No. 4 gave a study of the influence of elasticity on perfect gears. This present report will deal with the influence of errors in the tooth profiles on the acceleration loads.

II—INFLUENCE OF ERRORS ON ACCELERATION LOADS

Errors on gear-tooth profiles, assuming a constant velocity of the driving member, act to accelerate and decelerate the driven member. This varying velocity of the driven member results in a varying load on the gear teeth, the amount of this variation depending upon the masses of the revolving parts, the nature and extent of the errors, and the velocity of the gears. If the gears were made of rigid materials, the maximum acceleration load would vary as the square of the velocity. With elastic materials, however, as the acceleration load increases, the deformation of the tooth profiles will also increase, and this deformation will increase most where the acceleration load is greatest and hence tend to reduce this load by reducing the acceleration. Under these conditions the force of the acceleration would be given by Equation [I-17] from the previous study, which is as follows:

$$f_a = \frac{C_1 V^2 C_2}{C_1 V^2 + C_2} \qquad [1]$$

where f_a = force of acceleration
$f_1 = C_1 V^2$ is the equation of the parabola representing the reactions of rigid solids, and
$f_2 = C_2$ is the equation of the horizontal asymptote which represents the load required to deform the teeth enough to eliminate acceleration.

We shall first direct our attention to the equation of the parabola. In order to establish this equation, we must first determine the amount of the effective error and the distance in which it acts.

It appears from a study of the diagrams or charts which measure the accuracy of the gears that the effective error seems to act while the load is being transferred from one pair of gear-tooth profiles to the next pair. The errors may be on the tooth profiles or in the spacing, but their influence seems always to be most

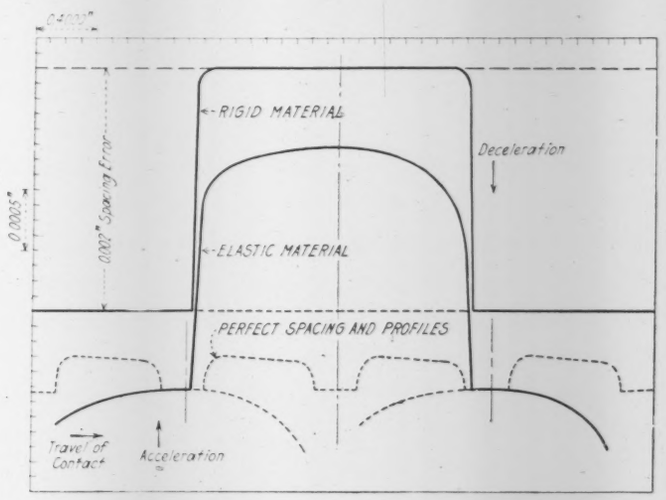

FIG. 1 DIAGRAMS OF EFFECTIVE ERRORS
(3-D.P., 20-deg., 18-tooth pinion and 48-tooth gear; 1000 lb. static load.)

during the transfer of the load from one pair of teeth to the next pair.

When a definite error is present, the engaging pair of teeth will tend to take over the entire load. When the error is greater than the static deformation under load, one pair of teeth only will carry the load except during the time that the load is being transferred. A chart of the static deformation of such gears would show an effective error approximately equal to the error.

A diagram of the effective error that would exist on a 3-D.P. steel gear and pinion under static conditions with an applied load of 1000 lb. is shown in Fig. 1. These gears are the same as those shown in Figs. 2 and 3 of the first study except that one tooth

has a spacing error of 0.002 in. A dotted line shows the effective errors of perfect gears under the same static conditions. A full line shows the effective error with rigid materials, while the lower full line shows the effective error under static conditions with the elastic materials.

Due to the fact that the engaging tooth takes over the load at the tip where the deflection is greatest, the actual effective error here is slightly less than the spacing error. This difference is equal to about one-half the static deflection at the pitch line. Thus as the applied load is increased, the acceleration conditions will be modified slightly. The full line in Fig. 1 which represents

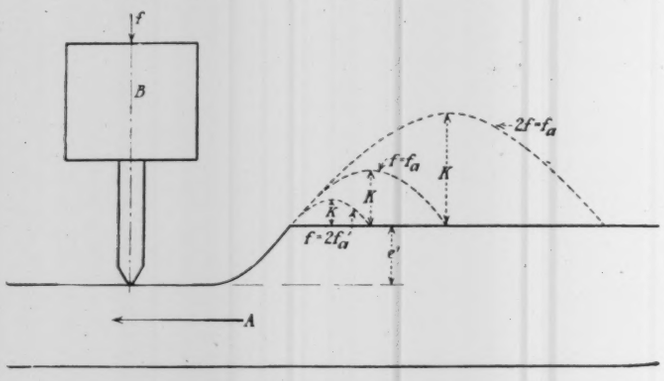

FIG. 2 SEPARATION OF RIGID BODIES

the effective error with rigid materials also represents the effective error with a zero applied load. In this case when

e' = actual spacing error
e = effective error
d_t = static deformation at the pitch line

we have Equation [I-26] as follows:

$$d_t = P\left(\frac{E_1z_1 + E_2z_2}{E_1z_1E_2z_2}\right) \quad [2]$$

and Equation [I-25] as follows:

$$z = \frac{1}{0.242/y + 7.25} \quad [3]$$

where E_1 & E_2 = modulus of elasticity of material
P = applied load
z = elasticity form factor
y = Lewis form factor.

Then

$$e = e' - \frac{d_t}{2} \quad [4]$$

Assuming these conditions of static deformation as rigid, the next step is to determine the distance in which this effective error will act. This distance may depend upon the nature of the error. On the other hand, it may be that these conditions approach some constant. This must be determined by actual experiment. For spacing errors alone, it may be that the approximation given by Equation [I-27] will be sufficiently accurate. Another approximation which will give almost identical results for the small errors involved is the following:

$$D = \frac{R_1R_2}{R_1 + R_2}\sqrt{\frac{2e}{0.75p}} \quad [5]$$

where D = distance in which error acts, inches
R_1 = pitch radius of pinion, inches
R_2 = pitch radius of gear, inches
e = effective error, inches
p = circular pitch, inches.

The equation which would represent the acceleration load under the assumed rigid conditions would be similar to Equation [I-16] as follows:

$$f_1 = \frac{meV^2}{150D^2} \quad [6]$$

where f_1 = acceleration load on rigid materials
m = effective mass at pitch line
V = pitch-line velocity in feet per minute.

The relationship between D and e is such that

$$\frac{e}{D^2} = 0.375p\left(\frac{R_1 + R_2}{R_1R_2}\right)^2$$

whence

$$f_1 = 0.0025p\left(\frac{R_1 + R_2}{R_1R_2}\right)^2 mV^2 \quad [7]$$

In other words, the parabola representing the reactions of rigid solids for a given pair of gears, with an error such as a spacing error, may be constant regardless of the extent of the error. It should be kept in mind that this relationship has been established by means of approximations, yet it is believed that these approximations are reasonably accurate. It is quite possible, therefore, that any variations in this parabola are so slight that for all practical purposes they may be ignored.

We shall now turn our attention to the equation of the horizontal asymptote. This will represent the load required to deform the tooth an amount equal to the static deformation plus the error in spacing. Hence.

$$f_2 = f\left(\frac{e' + d_t}{d_t}\right) = f\left(\frac{e'}{d_t} + 1\right) \quad [8]$$

where f = applied load.

Equation [1] will then become

$$f_a = \frac{f_1f_2}{f_1 + f_2} \quad [9]$$

Equations [7] and [8] could be combined into Equation [9], but as they could not be simplified, nothing would be gained by so doing.

In the foregoing analysis the assumption was first made that the velocity of the driving member is constant and that all variation in velocity takes place on the driven member. Actually this variation in velocity will be divided between the two members, depending upon their relative effective masses. Thus if these masses are equal, the variation in velocity will be divided equally between them, and the resultant effective mass influence on the acceleration load will be equal to one-half of either equal mass. Thus when

m = resultant mass influence on acceleration load
m_1 = effective mass of pinion at pitch line
m_2 = effective mass of gear at pitch line,

$$m = \frac{m_1m_2}{m_1 + m_2} \quad [10]$$

SEPARATION OF TEETH BECAUSE OF ACCELERATION

At the instant that the second pair of tooth profiles are carrying the load and the acceleration force has ceased, the masses are moving apart from each other because of the difference in velocity imparted to them by the accelerating force. This relative movement apart is resisted by the applied load f. If the materials were rigid, we should have the conditions shown in Fig. 2. The slide A is moving in the direction shown by the arrow. The force f is holding mass B against this slide. When the mass B is moving up the cam portion, the accelerating force f_a is acting through the vertical distance e'. The additional work done by this cam is equal to $f_a \times e'$. The distance that the body B will part from the slide will depend upon the amount of the force f. This force will act through the distance k to absorb the additional work of acceleration, hence

$$f_a \times e' = f \times k \quad [11]$$

Thus when f is equal to f_a, these rigid bodies will separate a distance equal to e'. When f is double f_a, the distance k will be one-half the distance e', etc. This separation is shown by dotted lines in Fig. 2.

With elastic bodies, however, the acceleration force in addition

to imparting a certain difference in velocity between the masses, will also cause additional deformation. Under these conditions, assuming the slide A only as elastic, we should have the conditions shown in Fig. 3. In this case the work which is stored in the deformed material will also tend to separate the bodies in addition to the separation force of the acceleration, while the applied load will act as before to hold them together. Thus when

f = applied load
f_a = acceleration load
d_t = deformation due to the applied load
d_m = maximum deformation due to the total load
e' = error or rise on undeformed profile,

the additional work done by the acceleration force will be equal to $f_a(e' - d_m + d_t)$. The additional work stored in the deformed profile will be equal to $\frac{1}{2}f_a(d_m - d_t)$. The applied load f will act through the distance $(k + d_t)$ to absorb this additional work, whence

$$f_a(e' - d_m + d_t) + \tfrac{1}{2}f_a(d_m - d_t) = f(k + d_t)$$

whence

$$k = \frac{f_a}{f}\left(e' - \frac{d_m}{2} - \frac{d_t}{2}\right) - d_t \qquad [12]$$

$$d_m = \left(\frac{f_a}{f} + 1\right) d_t$$

whence

$$k = \frac{f_a}{f} e' - \left(\frac{f_a}{f}\right)^2 \times \frac{d_t}{2} - dt \qquad [13]$$

When $f_a = 0$ (static conditions),

$$k = -d_t$$

When $f_a = f$,

$$k = e' - \frac{3}{2} d_t$$

It will be noted that the influence of elasticity on this separation is to reduce its amount.

When we attempt to derive similar equations for these separation conditions on gear teeth we find that we have a much more complex problem to contend with, because at one instant two pairs of mating teeth may be sharing the load, while at another instant but a single pair will be in action. Although for relatively large tooth errors we can probably assume that the conditions are practically identical to those shown in Fig. 3 because a single pair of teeth will take over practically all of the load, yet when the error is small or when the deformation becomes great enough, two pairs of teeth will be sharing these loads, and the exact distribution of these loads will always be uncertain.

Considering first these conditions on perfect gears, we have two pairs of teeth in contact during this acceleration. In this case the effective error is equal to $\frac{1}{2}d_t$, where d_t is the static deformation caused by the applied load acting on a single pair of teeth. Whence the additional work done by the acceleration force will be equal to $\frac{1}{2}f_a(d_t - d_m + d_t) = f_a(d_t - \frac{1}{2}d_m)$. The additional work stored in the deformed profiles will be equal to $\frac{1}{2}f_a \times (d_m - d_t)$. The applied load f will act through the distance $(k + \frac{1}{2}d_t)$ to absorb this additional work, whence

$$f_a(d_t - \tfrac{1}{2}d_m) + \tfrac{1}{2}f_a(d_m - d_t) = f(k + \tfrac{1}{2}d_t)$$

whence

$$k = \left(\frac{f_a}{f} \times \tfrac{1}{2} d_t\right) - \tfrac{1}{2}d_t = \tfrac{1}{2}d_t\left(\frac{f_a}{f} - 1\right) \qquad [14]$$

When $f_a = f$, $k = 0$, and when $f_a = 0$ (static conditions), $k = -\frac{1}{2}d_t$, which is the static deformation at this part of the profile. As we have seen before, the acceleration load on perfect gears can never be larger than the applied load, so that all values of k under these conditions will be minus, with zero as its maximum limit.

When we introduce a small error on the gear-tooth profiles and two teeth are sharing the acceleration load, we shall assume that they share it equally. Thus when e = effective error on gear tooth profile, the additional work done by the acceleration force will be equal to $f_a(e - \frac{1}{2}d_m + \frac{1}{2}d_t)$. The additional work stored in the deformed profiles will be equal to $f_a(\frac{1}{2}d_m - \frac{1}{2}d_t)$. The applied load f will act through the distance $(k + \frac{1}{2}d_t)$ to absorb this additional work, whence

$$f_a(e - \tfrac{1}{2}d_m + \tfrac{1}{2}d_t) + f_a(\tfrac{1}{2}d_m - \tfrac{1}{2}d_t) = f(k + \tfrac{1}{2}d_t)$$

whence

$$k = \frac{f_a}{f} e - \tfrac{1}{2}d_t$$

When e' = actual error on gear-tooth profiles, under the foregoing conditions, $e = e' + \frac{1}{2}d_t$, whence

$$k = \frac{f_a}{f}(e' + \tfrac{1}{2}d_t) - \tfrac{1}{2}d_t \qquad [15]$$

Equation [15] is manifestly wrong because when $f_a = f$, the value of k would be the same as that for two rigid solids under the same conditions. The assumption that two teeth could share the load equally when a definite error is present must therefore be rejected. The tooth with the error would obviously carry the larger part of the load, which would introduce a different value for the amount of deformation.

When the error is larger than the maximum deformation, one

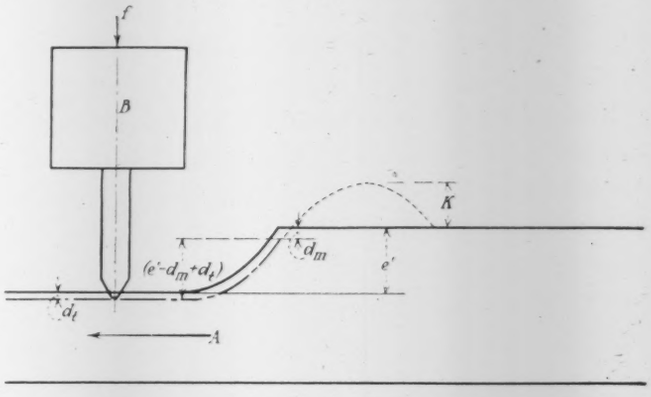

FIG. 3 SEPARATION OF ELASTIC BODIES

pair of teeth only will carry the entire load, and Equation [13] should give a reasonably accurate value of k. As the applied load or the acceleration load becomes greater so that the effective error is less than the deformation, a second pair of teeth will begin to share the load and the value of k will be different from that given by Equation [13]. At infinite speed the masses will move uniformly, and all variations in tooth pressure will be caused by the varying deformation of the material necessary to permit these masses to move uniformly. Under these conditions, according to the assumption leading to the derivation of Equation [8],

$$\frac{f_a}{f} = \left(\frac{e'}{d_t} + 1\right) \qquad [16]$$

Transposing Equation [13] to solve for f_a/f, we have

$$\frac{f_a}{f} = \frac{e' + \sqrt{(e')^2 - 2d_t(d_t - k)}}{d_t} \qquad [17]$$

Equations [16] and [17] are not comparable because under the conditions existing at infinite speed (Equation [16]) two pairs of teeth will be sharing the load, while Equation [17] holds true only when a single pair of teeth are engaged. Under the conditions of infinite speed it is possible that the gear-tooth profiles never lose actual contact, so that the maximum value of k would be zero. This value of k can probably be safely assumed as an asymptote. On all actual tests the probability is that the errors are large enough so that but a single pair of teeth are in contact at the instant of separation. In this case Equations [13] and [17] may be used in the analysis of the test results.

INFLUENCE OF NORMAL PITCH ERRORS

Fig. 4 shows the effective errors on the same gears as before when

a normal pitch error of minus 0.001 in. exists on the driving member. This diagram represents the conditions that are present when the normal pitch of the driving member is shorter than that of the driven member. Here as with spacing errors the load is carried by a single pair of teeth except while it is being transferred from one pair of teeth to the next pair. Assuming a constant velocity of the driving pinion, the driven gear falls back slowly during each tooth mesh and is advanced again rapidly when the contact shifts from tooth to tooth.

As in Fig. 1, in Fig. 4 one full line represents the effective error with rigid materials, or with zero applied load, under static conditions, while the lower full line represents the effective error with elastic materials under a static load of 1000 lb.

The effective error on the deformed tooth profiles here is also slightly less than on rigid materials. This error is less by very close to one-half the static deformation at the pitch line, so that

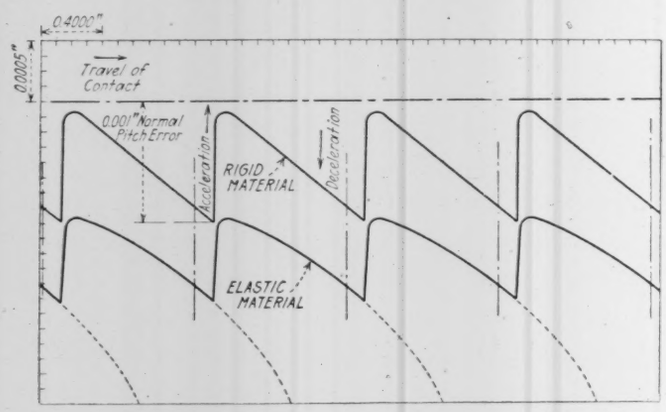

FIG. 4 DIAGRAM OF EFFECTIVE ERROR
(3-D.P., 20-deg. 18-tooth pinion and 48-tooth gear; 1000 lb. static load.)

the foregoing analysis of the effect of spacing errors on the acceleration load and separation of the tooth profiles holds substantially true here also.

The foregoing analysis is based on the assumption that uniform accelerations result from deformations and positive errors. This assumption is probably seldom exactly true, yet the influence of the elasticity of the materials will tend to make the accelerations approach this uniform condition.

As a matter of fact, any effective errors in the tooth profiles which would cause a sudden change in velocity may act practically the same as impacts, or suddenly applied loads, and the intensity of the loads thus set up possibly may be analyzed best as impact loads. The results on the Lewis gear-testing machine give the load required to hold the teeth in sufficiently close contact to maintain the electrical circuit through them. When this contact is broken, some definite space exists between the tooth profiles. Under static conditions with hardened and ground steel gears, this space is very close to two thousandths of an inch. Under these conditions a current of about 8 volts is registered on the voltmeter with full contact of the tooth profiles. In operation at the higher speeds the voltmeter registers from one to two volts. Part of this loss of voltage is probably due to the ball-bearing commutators and part to the imperfect contact of the gear teeth with each other. It is probable that with this lower voltage, the space through which the arc will jump is materially reduced. Static tests will be made to investigate this feature.

With cast-iron gears under static conditions, the distance the teeth must be separated to break the circuit is only about one-tenth of a thousandth of an inch with the full voltage of 8 volts. Thus the current seems to jump a much wider gap between polished steel surfaces than that between cast-iron ones. This accounts for the pitting or burning which has been experienced when running tests with the hardened and ground steel gears. It also accounts for the difficulty of obtaining satisfactory test readings on cast-iron gears at the higher velocities.

The foregoing analysis has been made in the attempt to establish some usable relationship between the test load on the machine, which is the transmitted load, the acceleration load, and the maximum space between the tooth profiles which has been designated as k, in order to be able to obtain some measure of the impact loads. The next step in this study is the consideration of impact loads on elastic materials. These impact loads are unquestionably the maximum and therefore the critical ones. We shall then try to determine whether or not the experimental results obtained by tests support these analyses. Needless to say, all of the many assumptions which are made in these analyses must be proved by the results of actual tests before they can be accepted as substantially true.

Scientific Research and Its Application to Industry

AT THE opening technical session of the spring sectional meeting of the American Society for Steel Treating, T. McLean Jasper, director of research, A. O. Smith Corporation, Milwaukee, read a paper on Scientific Research and Its Application to Industry. The perusal of work done by previous investigators and even, in some instances, the repetition of research work, were suggested by Mr. Jasper as aids to establishing a proper background for the successful termination of true research as distinguished from ordinary tests. A brief abstract of Mr. Jasper's paper follows:

Often undesired results in the application of research are brought about by the fact that it is attempted before the work of the laboratory, or before study is sufficiently completed to draw a sane and unbiased conclusion. Sometimes the cause of unsuccessful application of research is the result of biased data from which unsound conclusions have been drawn.

The production of scientific research is occasionally confused with the making of tests. Research involves much more than the making of tests. It involves the logical design of experiments which, when completed, will give a "yes" or "no" answer to a definite question. It involves (1) research leading to new discoveries, (2) research leading to new and more economical methods of producing old products, and (3) researches tending to supplement and improve old processes and products.

Of the three divisions outlined above, the first generally requires a higher degree of skill and application than the other two because less precedent is available on which dependence can be put for help. In the other two divisions the skill required in general decreases in the order named.

Too much stress cannot be placed on going through, as a preliminary, the work of other reliable investigators on similar or analogous problems. Such publications as the proceedings of the Royal Society, the various national academies of science, and the proceedings of professional societies, and such publications as the *Philosophical Magazine* and other technical papers and magazines, are particularly helpful, most of which are readily available in the various libraries in the country. The result of this preliminary study (which relatively few research workers are willing to carry through) creates an atmosphere which is congenial to developing a sense of feel and a sense of proportion for the thing in mind.

Assuming for the moment that the problem has been successfully solved, the next thing to do is to present the work in such a manner that it will be readily understood by those who can best use it. Too few workers engaged upon research realize the importance of this essential. In order that the results of successful investigation can be used in industry it frequently involves handing it over to an entirely separate organization for its successful application.

There are three general outlying important points which need to be considered in the question of applying changes to manufacturing processes, i.e., (1) a consideration of its effect on the existing economic situation, (2) a consideration of its cost, and (3) a consideration of its application with the class of workers which are involved in the change.

To draw attention to these factors may seem unnecessary to many, yet the fact remains that the lack of a full consideration of them has resulted repeatedly in lack of success and the unremunerative expenditure of much money. Case after case can be cited in which the above statements stand verified.—*The Iron Age*, May 26, 1927, p. 1535.

Ideal Gas-Engine Cycles

By ROBERT C. H. HECK,[1] NEW BRUNSWICK, N. J.

The purpose of this paper is to promote the use, with the internal-combustion engine, of the ideal performance as a standard of comparison, by making that standard almost as readily convenient as is the Mollier chart for the steam engine. This purpose is accomplished in two steps: (a) developing a working chart that combines in one the temperature-entropy diagram and a diagram of internal energy and total heat on entropy, in which any horizontal line is a line of constant temperature, constant energy, and constant total heat; and (b) showing by trial upon representative examples that this chart, based on the properties of an average gas mixture, gives essentially correct output for any working mixture within the range of gas-engine practice.

Avoiding the common type of chart, with its confusing multiplicity of curves, the present scheme employs the principle of moving curves, carried by templets. The chart carries one curve of constant-volume and one of constant-pressure heating, with horizontal scales of volume and pressure and vertical scales of energy and total heat or "enthalpy." These are so disposed that any problem can be solved by construction upon a separate working sheet, laid down across the chart, which will serve as a permanent record of the solution.

Important features of the paper are found in Figs. 2 and 3 on the specific heats of working mixtures, Figs. 4 and 5 for the general idea of chart and method, in the summary of the Otto-cycle determination, and in that of the Diesel-cycle determination.

EVERY heat engine has two principal efficiencies: the absolute efficiency e_a, which is the ratio of the heat converted to the heat received; and the relative efficiency e_r, which is the ratio of the actual to the ideal or best possible performance. To know e_r it is necessary to find the value of ideal efficiency e_i by theoretical deduction.

If q be the heat input per unit of working medium and w the work output, expressed in heat units and distinguished as actual w_a and ideal w_i, then

$$e_a = \frac{w_a}{q}; \quad e_i = \frac{w_i}{q}; \quad e_r = \frac{e_a}{e_i} = \frac{w_a}{w_i} \ldots\ldots [1]$$

For the steam engine the use of ideal performance as a standard of comparison has long been a routine matter; it is given by the Rankine or Clausius cycle, and is put into convenient practical form by the Mollier chart of total heat on entropy. With the gas engine the use of a similar standard has been retarded for two reasons: to calculate the dimensions of the ideal cycle is a difficult and laborious task, and with each different composition of gaseous mixture a different set of data enters the calculation. This paper presents a convenient and flexible method for getting numerical results, and shows how nearly representative are the results from a single, average gas composition.

Symbols, Units, Ratios

In the practical thermodynamics of the internal-combustion engine three systems of measurement are encountered, namely,

The English-fahrenheit system, symbol *ef*
The English-centigrade system, symbol *ec*
The metric-centigrade system, symbol *mc*.

Temperature. The 1.8 ratio of the fahrenheit to the centigrade measurement of a temperature range holds also for heat or energy quantities per unit of mass.

t = thermometer reading; $\quad T = t + 459.6$ (fahr.)
T = absolute temperature; $\quad T = t + 273.1$ (cent.)

Pressure. Engineering pressure p and thermodynamic pressure P.

p in lb. per sq. in. (Eng.) or kg. per sq. cm. (met.)
P in lb. per sq. ft. (Eng.) or kg. per sq. m. (met.)

$P = 144p$ (Eng.) $\qquad P = 10{,}000p$ (met.)
$p_e = 14.223p_m \qquad P_m = 4.882P_e$

Mass. The relative size of the pound and kilogram does not appear directly, because values of volume, energy, and entropy are stated per unit of mass; but the ratio 2.2046 enters into the relations between more complex units and ratios.

Volume. Specific volume v is the volume of the unit mass, at any particular state as defined by pressure and temperature,

v in cu. ft. per lb. (Eng.) or cu. m. per kg. (met.),

$$v_e = 16.02v_m$$

In the characteristic equation or equation of state of any gas (for its ideal behavior)

$$pv = bT \quad \text{or} \quad Pv = BT \ldots\ldots [2]$$

the gas constant b or B has relative values as follows:

$B = 144b$ (Eng.) $\qquad B = 10{,}000b$ (met.)
$b_{ef} = 126.57b_{mc} \qquad b_{ec} = 1.8b_{ef} = 227.83b_{mc}$
$B_{ef} = 1.8227B_{mc} \qquad B_{ec} = 1.8B_{ef} = 3.281B_{mc}$

The general gas equation is

$$pv = \frac{r}{m}T \quad \text{or} \quad Pv = \frac{R}{m}T \ldots\ldots [3]$$

in which r or R is the general gas constant and m is the molecular weight of the particular gas. The absolute values are

		ef	*ec*	*mc*
With engineering pressure p,	$r =$	10.732	19.30	0.0847
With thermodynamic pressure P,	$R =$	1544	2779	847.1

Putting $V = mv$ in Equation [3] gives

$$Pmv = PV = RT \ldots\ldots [4]$$

which is the equation of state for a mol of any gas. (The mol is the quantity of gas having a mass of m units.) Barring departures from ideal behavior the volume of the mol is the same for all gases, at the same pressure and temperature.

The product Pv implies the work of a constant-pressure expansion at P from zero volume to v. Then Pv is directly in work units, either foot-pounds or meter-kilograms; but pv is in units of 144 ft-lb. or 10,000 m-kg.

Pressure-volume work is converted to heat units by the factor A, the heat equivalent of work, this A being the reciprocal of J, the mechanical equivalent of heat. Besides the B.t.u. and the kilogram-calorie there is a third heat unit, the centigrade heat unit or c.h.u., which is the heat required to raise 1 lb. of water 1 deg. cent. The three values of J are

	ef	*ec*	*mc*
Mechanical equivalent $J =$	777.6	1400	426.6

With engineering pressure p in product pv, the effective value of factor A is not simply $1/J$ but is either $144/J$ or $10{,}000/J$; thus

	ef	*ec*	*mc*
Effective A with pv, $A' =$	1/5.4	1/9.72	23.44

For heat, energy, and entropy quantities the symbols, all per unit of mass, are

q = heat imparted or received
u = change of internal energy
i = total heat or "enthalpy," the heat added under constant pressure from a chosen initial temperature
ϕ = entropy change or entropy from a chosen origin.

Entropy, a pure ratio, has the same value in all systems.

For specific heat, the symbols c_p and c_v are used for common values per unit mass and the symbols C_p and C_v for heat rate

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[1] Professor of Mechanical Engineering, Rutgers University. Mem. A.S.M.E.

Contributed by Oil and Gas Power Division and presented at the Annual Meeting, New York, December 6 to 9, 1926, of The American Society of Mechanical Engineers. Abridged.

per mol per degree. (The values are, of course, independent of the system.) For the ideal gas the difference $c_p - c_v$ is equal to AB, which may be called the specific heat of outer work under constant pressure. Between C_p and C_v there is the constant difference AR which has the value 1.986 when found with any pair of the R and J values above, or in any system.

SIMPLIFYING ASSUMPTIONS

First, the gaseous medium, both before and after combustion, is assumed to follow volumetrically the simple ideal law $pv = bT$, which will be more nearly true for the mixture than for some of the separate ingredients. Departures from this form are due to two causes, namely, molecular volume and the contractive influence of molecular attractions. The first has a small effect directly proportional to pressure, the second is stronger toward the region of liquefaction. Over the range of exhibited observations air shows a contraction that persists but is rapidly decreasing (with rise of

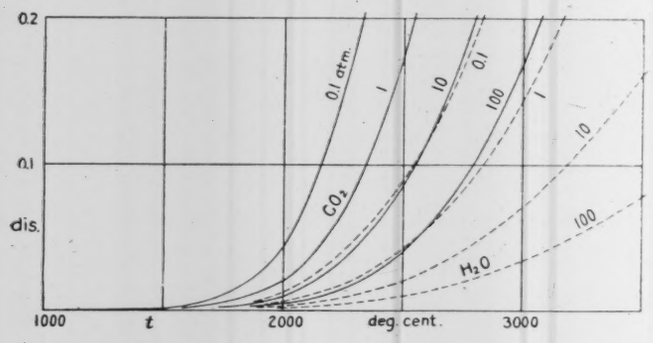

FIG. 1 NERNST'S DATA AS TO THE DISSOCIATION OF CO_2 AND H_2O

temperature), while in the case of hydrogen this effect has disappeared. The latter type of behavior is expressed by the equation

$$p(v - w) = bT \quad \text{or} \quad pv = bT + pw \qquad [5]$$

in which w is a small constant representing the combined volume of the molecules.

Considering the comparatively low pressure range of gas-engine conditions, and the fact that high pressures are associated with high temperatures, it is entirely proper to disregard the very small departures from the ideal law of Equation [2]. With equal propriety the small actual departures from Avogadro's law, as expressed by Equation [3], may be ignored; and in that formula it will be sufficiently accurate to use the simple integral values of molecular weight m.

A consequence of essential conformity to either Equation [2] or Equation [5] is that the specific heats c_p and c_v do not vary with pressure along an isotherm, however they may increase with temperature.

The second major assumption is that combustion is complete as a simple overall operation, and need not be considered in detail. This is carried so far as to consider even dissociation to be an element of actual as against ideal action. That is, we imagine a limiting combustion in which the total heating value becomes immediately active and raises the final gaseous products to the maximum temperature derivable from specific-heat relationships. Of course, heat insulation is assumed to be perfect in the ideal case.

About dissociation—as an item of chemical equilibrium at high temperatures or of retarded combination—there is little quantitative information. Fig. 1 shows graphically the lower range of the small Nernst table which is generally quoted in reference books. The curves are lines of constant pressure, on a base of temperature, and the ordinate is the proportion or fraction of ultimate CO_2 or H_2O that remains separated into CO and O_2 or H_2 and O_2.

SPECIFIC HEAT OF GASES

Interest attaches to the molecular specific heat C and to the pound specific heat c, for constant pressure as C_p or c_p and for constant volume as C_v or c_v. For a gas of ideal volumetric behavior the constant-volume specific heat has a general meaning as the rate of change of internal energy u with temperature t or T in any action. These ratios of heat added to temperature change all vary with the temperature in a manner that can be accurately enough represented by the equations

$$C = A + BT + GT^2 \qquad [6a]$$

$$c = \alpha + \beta T + \gamma T^2 \qquad [6b]$$

Upon this subject the readiest sources of information are the book The Specific Heat of Gases, by Partington and Shilling,[2] and an Investigation of the Maximum Temperature Attainable in the Combustion of Gaseous and Liquid Fuels, by Goodenough and Felbeck.[3]

From the Partington and Shilling formulas correctly converted from t to T as base, the coefficients for Equation [6a] are as follows:

		A_p	A_v	B_c	B_f	G_c	G_f
(a)	Air, N_2, O_2, CO	6.933	4.947	0	0	0.0₆31	0.0₆957
(b)	CO_2	7.382	5.396	0.00506	0.00281	−0.0₆102	−0.0₆312
(c)	H_2O	7.12	5.13	0.00256	0.00144	0.0₆405	0.0₆125

Coefficient A has different values for constant-pressure and constant-volume heating, but is the same with either temperature scale. The difference between A_p and A_v, or between the mol specific heats C_p and C_v, is 1.986 or 1.99. The two values of B and G are for the two temperature scales, with the ratios 1.8 and 3.24. Hydrogen is not stated here because that gas appears only as an ingredient in combustible mixtures, which are considered later. An important fact is the strong influence that the presence of H_2O will have in raising the mean specific heat of a mixture above that for air alone. It should be noted that in this paper the "mean" specific heat is always the average for a mixture of gases, not the mean value over some temperature range.

The scanty information available as to the specific heat of the hydrocarbon gases and vapors leads to straight-line relationships of the form

$$c_v = \alpha + \beta T \qquad [7]$$

The values of the coefficients are given later.

EXAMPLES FOR STUDY

The special problem of the paper cannot be solved by direct analysis, but must be handled by the method of trial solution, applied to representative examples. Covering the whole field of gas-engine practice, ten such examples have been chosen, namely, six fuel-gas mixtures, two limiting oil compositions, and the two common alcohols—the last taken to be pure fuel, or without admixture of water.

For the fuel gases the examples given by Goodenough[4] are used, with one changed (producer gas) and one added (retort or coke-

TABLE 1 FUEL-GAS COMPOSITIONS BY VOLUME, ETC.

Gas No.		1	2	3	4	5	6
Symbol		B. F.	Prod.	Wat.	C. W.	Nat.	Ret.
Hydrogen	H_2	0.04	0.13	0.50	0.30	0.02	0.50
Carbon monoxide	CO	0.27	0.25	0.45	0.27	0.01	0.07
Methane	CH_4	....	0.03	....	0.25	0.93	0.32
Ethylene	C_2H_4	....	....	....	0.15	....	0.04
Carbon dioxide	CO_2	0.10	0.05	0.03	0.01	0.01	0.03
Nitrogen	N_2	0.59	0.54	0.02	0.02	0.03	0.04
Total		1.00	1.00	1.00	1.00	1.00	1.00
Mean molecular weight m_m		28.56	25.06	15.48	17.36	16.48	11.64
Specific volume relative to air		1.014	1.157	1.871	1.668	1.758	2.489

oven gas). Shown as to volume composition and relative volume by Table 1, these gases are as follows:

1 Blast-furnace gas, the leanest of engine fuels
2 Producer gas, high in nitrogen and with but little hydrocarbon whatever the coal used
3 Water gas, of high-grade type
4 Carbureted water gas, the standard illuminant
5 Natural gas, nearly all hydrocarbon
6 Retort or coke-oven gas.

These are arranged in the order of increasing hydrogen content or of H_2O content in the products of combustion.

The mineral oils are adequately represented by two weight compositions for the pure combustible, namely,

[2] Benn, London, 1924.

[3] Bulletin 139, March, 1924, Engineering Experiment Station, University of Illinois.

[4] See Principles of Thermodynamics, p. 297.

7 Heaviest oils, $H_2 = 0.12$, $C = 0.88$
8 Lighter oils, gasolines, $H_2 = 0.16$, $C = 0.84$.

As regards hydrogen content these are extreme compositions, showing a bigger range of variation than is likely to be encountered; especially, 12 per cent of H_2 is less than is to be found in oil analyses.

The alcohols are:

9 Ethyl or grain alcohol, C_2H_6O
10 Methyl or wood alcohol, CH_4O.

Besides composition by volume, Table 1 gives the mean molecular weight of the fuel gas and the ratio of its specific volume to that of air. Since volumes are inversely as molecular weights, this ratio is equal to m_a/m_m or $28.97/m_m$. The first two gases are near to air, but the others are very much lighter because of the content of H_2 and CH_4.

In Table 2 the ten examples are brought to weight measurement, with the pound of fuel as base, and are compared as to combustible content, air requirement and heating value. The first three

TABLE 2 COMBUSTIBLE, AIR RATIOS, HEATING VALUES

	1	2	3	4	5	6	7	8	9
		C in				Per lb. of fuel		Heating value	
Fuel	H_2	hydro carbons	C in CO	Sum	Comb.	Air	Mixture	Fuel	Mixture
1 B.F.	0.003	...	0.113	0.116	0.268	0.860	1.860	1,303	701
2 Prod.	0.015	0.014	0.120	0.149	0.309	1.585	2.585	2,183	844
3 Wat.	0.065	...	0.349	0.414	0.879	4.867	5.867	6,970	1188
4 C.W.	0.127	0.380	0.187	0.694	0.943	11.29	12.29	13,660	1112
5 Nat.	0.228	0.676	0.007	0.911	0.921	17.11	18.11	19,710	1038
6 Ret.	0.210	0.413	0.072	0.695	0.791	14.24	15.24	16,780	1100
7 Oil	0.120	0.880	...	1.000	1.000	16.48	17.48	18,600	1058
8 Gaso.	0.160	0.840	...	1.000	1.000	17.54	18.54	18,500	1000
9 Eth.	0.130	0.522	...	0.652	1.000	13.56	14.56	11,200	769
10 Meth.	0.125	0.375	...	0.500	1.000	9.75	10.75	8,400	781

columns give the weight of hydrogen, of carbon in hydrocarbons, and of carbon in CO. These are summed for the combustible elements in column 4, then the total combustible in column 5 is greater by the oxygen in CO or in the alcohol. The air required per pound of fuel is given in column 6, calculated with 15 per cent excess for the gases and oils and with 50 per cent excess for the alcohols. Adding the one pound of fuel we get the total weight of mixture in column 7. The heating values, per pound of fuel and per pound of mixture, are given in columns 8 and 9.[5]

The characteristics of the various gaseous media considered are further shown by Table 3. The first two columns give the volume of H_2O and CO_2 in one cubic foot of products, the next two the similar weights per pound of products. The rest of the table compares the precombustion mixture (subscript m) and the products of combustion (subscript p) as to mean molecular weight, volume relative to air, and outer-work rate AB under constant

TABLE 3 COMPARATIVE PROPORTIONS

	Column No. 1	2	3	4	5	6	7	8	9	10	11
	In products										
	Volume		Weight		Molecular weights		Relative volumes			Outer-work rate	
Fuel	H_2O	CO_2	H_2O	CO_2	m_m	m_p	v_m	v_p	v_p/v_m	AB_m	AB_p
1 B.F.	0.024	0.218	0.014	0.306	28.69	31.31	1.010	0.926	0.916	0.0693	0.0635
2 Pro.	0.087	0.159	0.053	0.224	26.83	29.60	1.080	0.979	0.907	0.0740	0.0671
3 Wat.	0.159	0.153	0.099	0.233	25.14	28.95	1.152	1.001	0.868	0.0790	0.0686
4 C.W.	0.147	0.111	0.093	0.171	27.36	28.40	1.059	1.020	0.963	0.0726	0.0700
5 Nat.	0.166	0.084	0.108	0.133	27.75	27.78	1.044	1.043	0.999	0.0716	0.0715
6 Ret.	0.189	0.077	0.124	0.124	26.29	27.45	1.098	1.055	0.958	0.0756	0.0724
7 Oil	0.100	0.122	0.062	0.185	...	29.06	...	0.996	...	...	0.0683
8 Gaso.	0.123	0.108	0.078	0.166	30.00	28.59	0.966	1.013	1.049	0.0663	0.0695
9 Ethl	0.198	0.050	0.131	0.081	29.60	28.34	0.979	1.022	1.045	0.0671	0.0701
10 Meth.	0.195	0.065	0.128	0.105	29.10	27.92	0.995	1.038	1.042	0.0683	0.0712

pressure. The ratio in column 9 indicates the shrinkage or swelling of specific volume that takes place during combustion. With all of the gas fuels there is a shrinkage, running as high as 13 per cent in one case; with the heavier liquid-fuel molecules there is an expansion running up to 5 per cent.[6]

Specific Heats of Working Mixtures

The specific heats of the combustible mixtures are of practical interest only over the comparatively short temperature range of the adiabatic compression in Otto-cycle engines, which will not run above 600 deg. cent. Straight lines that agree closely with the data of Partington and Shilling over this range may be represented by

$$C_v = A + BT \quad \ldots\ldots [8]$$

with coefficients as follows:

	Air, etc.	CO_2	H_2O	CH_4	C_2H_4
A	4.76	5.30	4.90	1.48	4.7
B	0.0_354	0.0038	0.0031	0.019	0.012

It must be remembered that, because of its small molecular weight, hydrogen has a very high "pound" specific heat; while CO_2, with a larger molecular weight, has a c_v not much above that of air.

Applied to the six examples of fuel-gas mixtures these coefficients for "mol" C_v's give the values of coefficients α and β for the mixtures listed in Table 4.[7]

TABLE 4 SPECIFIC HEATS OF MIXTURES

Fuel	α_v	AB	α_p	β
1	0.1670	0.0693	0.2363	0.0_4250
2	0.1764	0.0740	0.2504	0.0_4310
3	0.1895	0.0790	0.2685	0.0_4226
4	0.1683	0.0726	0.2409	0.0_4533
5	0.1625	0.0716	0.2341	0.0_4722
6	0.1752	0.0756	0.2508	0.0_4577

These are coefficients for use in the specific-heat equation $c_v = \alpha + \beta T$, with T in deg. cent., for combustible mixtures of the six gas fuels in Table 1, with 15 per cent excess air.

The mean of lines 2 to 6 in Fig. 2 is represented by the series of cross-marks on the ordinates. These lie in a straight line, the equation of which (with T in deg. cent.) is

$$c_v = 0.1745 + 0.00476T \quad \ldots [9]$$

From Table 3 the mean value of AB for these five examples is 0.0745, therefore the specific heat of the average mixture under constant pressure is

$$c_p = 0.2490 + 0.00476T \quad \ldots [10]$$

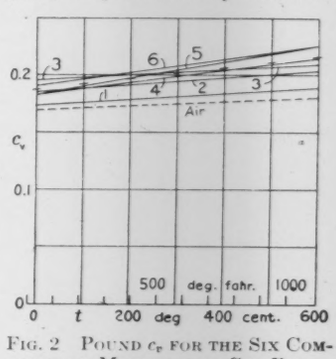

FIG. 2 POUND c_v FOR THE SIX COMBUSTIBLE MIXTURES OF GAS FUELS

In like manner the specific heats of the combustion products from the ten fuels in Table 1 are shown by the curves in Fig. 3. The numbering of these curves corresponds with that of the examples, except as to 9 and 10. The two alcohol products have specific heats so nearly the same that their curves are barely distinguishable when drawn with the scales of Fig. 3. Curve 9, which agrees with curve 6 over two-thirds its temperature range, is a mean for the two alcohols with 50 per cent excess air, and curve 10 is a similar mean with 15 per cent excess. Coefficients to be used in Equation [6b] with T in deg. cent. are given in Table 5.[7]

The mean value of c_v for fuels 2 to 8, omitting blast-furnace gas and alcohol, is shown by the black-dot "curve" in Fig. 3, for which the equation is

$$c_v = 0.1760 + 0.0_433T + 0.0_86T^2 \quad \ldots [11]$$

also with T in deg. cent. By a further averaging of these seven examples the mean value of AB or $c_p - c_v$ is found to be 0.0695. For reasons of convenience, and strongly justified as the scheme works out to final shape, the small departure from the dry-air value 0.0686 is disregarded and the latter is used. Then for the constant-pressure operation

$$c_p = 0.2446 + 0.0_433T + 0.0_86T^2 \quad \ldots [12]$$

TABLE 5 SPECIFIC HEATS ON THE PRODUCTS OF COMBUSTION

Fuel	α_v	AB	α_p	β	γ
1	0.1614	0.0635	0.2249	0.0_4372	0.0_8071
2	0.1700	0.0671	0.2371	0.0_4334	0.0_8396
3	0.1747	0.0686	0.2433	0.0_4410	0.0_8420
4	0.1770	0.0700	0.2470	0.0_4331	0.0_8623
5	0.1806	0.0715	0.2521	0.0_4310	0.0_8770
6	0.1829	0.0722	0.2551	0.0_4318	0.0_8820
7	0.1729	0.0683	0.2412	0.0_4302	0.0_8541
8	0.1721	0.0695	0.2416	0.0_4304	0.0_8623
9	0.1767	0.0700	0.2467	0.0_4267	0.0_8737
10	0.1793	0.0711	0.2504	0.0_4297	0.0_8776

[5] An example showing the calculation of these and following quantities is worked in the Appendix accompanying the complete paper, which latter is to appear in vol. 48 of Trans. A.S.M.E.

[6] This matter is more fully discussed in the Appendix.

[7] For discussion, curves, and method of deriving this table, see complete paper to appear in Trans. A.S.M.E., vol. 48.

The practical use of these specific-heat formulas is found in the calculation of entropy in constant-volume and in constant-pressure heating, of internal energy and of total heat. Changing to common logs by the factor 2.3026 and taking the state at 0 deg. cent. as the origin in each case, Equations [11] and [12] give

$$\phi_v = {}^{8}[9.60773] \log \frac{T}{273} + 0.0_4 33(T - 273) + 0.0_8 3(T^2 - 273^2) \quad [13]$$

$$\phi_p = \phi_v + [9.19854] \log \frac{T}{273} \quad [14]$$

$$u = 0.1760(T - 273) + 0.0_4 165(T^2 - 273^2) + 0.0_8 2(T^3 - 273^3) \quad [15]$$

$$i = u + 0.0686(T - 273) \quad [16]$$

An outstanding purpose of the paper is to obviate the necessity of calculating cycle dimensions by such relationships as

$$\alpha_p \log_e \frac{T}{T_1} + \beta(T - T_1) + {}^1/_2\gamma(T^2 - T_1^2) = AB \log_e \frac{p}{p_1} \quad [17]$$

or

$$\alpha_v \log_e \frac{T}{T_1} + \beta(T - T_1) + {}^1/_2\gamma(T^2 - T_1^2) = AB \log_e \frac{v_1}{v} \quad [18]$$

for the adiabatic operation and

$$q = \alpha(T - T_1) + {}^1/_2\beta(T^2 - T_1^2) + {}^1/_3\gamma(T^3 - T_1^3) \quad [19]$$

for the heating operations, with T as unknown quantity in each case.

Graphical Method for Cycle Dimensions

The general idea of the method now to be presented is to combine in one the temperature-entropy diagram and a diagram of energy or of total heat on entropy—the last being equivalent in effect to the Mollier chart for the steam engine. The graphical apparatus is shown by Fig. 4 and its use is illustrated by Fig. 5. At first we assume the simple condition of like properties, thermal and

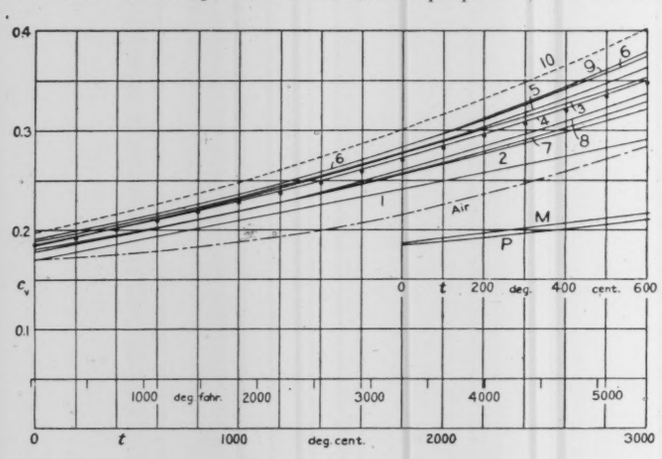

FIG. 3 POUND SPECIFIC HEAT c_v FOR THE PRODUCTS OF COMBUSTION OF THE TEN FUELS IN TABLE 1

volumetric, for the combustible mixture and the combustion products, using those of the average products as made definite in Equations [11] and [12]. After describing this scheme of solution, which meets the first main purpose of the paper as stated at the beginning, the effect of variations will be investigated and discussed.

The parts of Fig. 4 are as follows:

Curve V applies to constant-volume heating and curve P to constant-pressure heating. On the temperature base T at the left these show entropy ϕ_v or ϕ_p from Equations [13] and [14], to the scale along the horizontal base line. These curves are drawn in zero position, or with the zero of entropy at the standard point where $t = 0$ deg. cent. or 32 deg. fahr. and $p = 14.7$ lb., the latter value characterizing the P-curve. Then the V-curve corresponds to the specific volume 12.38 cu. ft. For any other volume or pressure the curve is simply shifted horizontally according to the scale marked V or P. An accurately cut templet of either curve is used instead of drawing a chart with a confusing multiplicity of curves.

Energy content u and total heat i, each per pound of medium

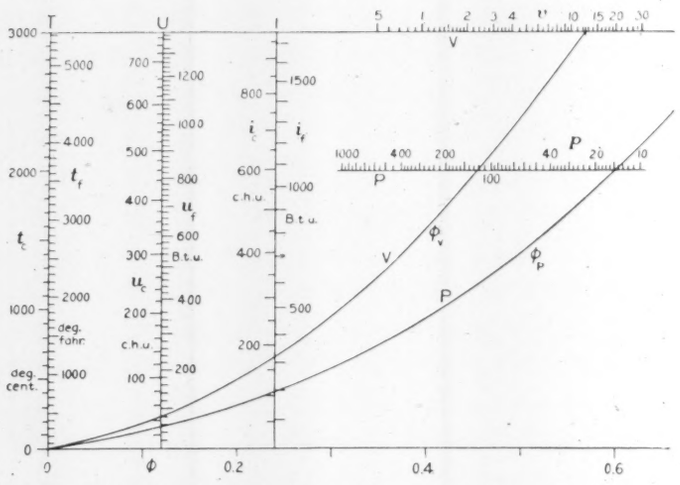

FIG. 4 GRAPHICAL APPARATUS FOR DETERMINING IDEAL GAS-ENGINE CYCLES

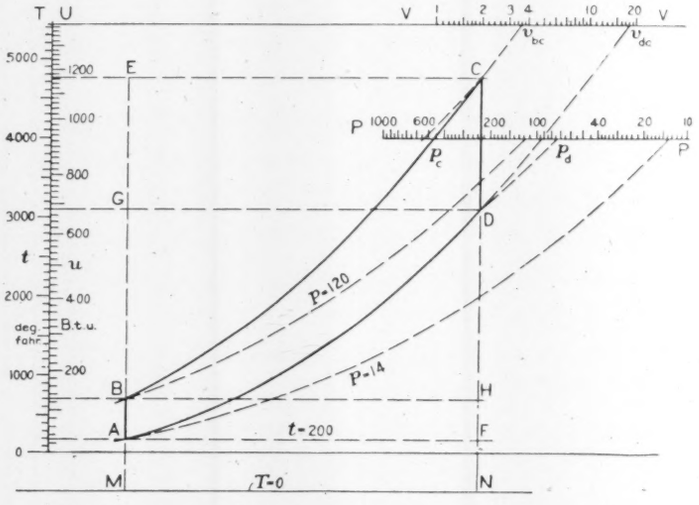

FIG. 5 ILLUSTRATIVE DETERMINATION OF AN OTTO CYCLE

and estimated from 0 deg. cent., are shown by the scales U and I. The fact that specific heat depends only upon temperature makes any horizontal line a line of constant u or of constant i.

Passage from one volume curve to another or from one pressure curve to another is most simply an operation of isothermal expansion or compression. That is, the horizontal entropy distance between any pair of curves is

$$_1\phi_2 = 2.3026\,AB \log \frac{v_2}{v_1} = 2.3026\,AB \log \frac{p_1}{p_2} \quad [20]$$

In Fig. 5 are reproduced only the needed vertical scales of t in deg. fahr. and u in B.t.u. Initial point A is located by drawing a pressure curve through $p_a = 14$ lb. and a temperature line at $t_a = 200$ deg. Adiabatic compression to $p = 120$ lb. is defined by the intersection of the vertical AB with the p_b-curve at that pressure.

On energy scale U set off now the height BE of the heat of combustion q, conceived simply as heat added in constant volume to the pound of gaseous medium. Through B and A draw the constant-volume curves BC and AD; intersection C fixes the state at the end of ideal combustion with no escape of heat, then vertical CD shows adiabatic expansion and fixes fourth point D, from which

[8] This notation uses the bracketed log as symbol for the antilog.

the gas is taken to return to state A by constant-volume cooling.

The work output is either the difference between adiabatic works CD and AB or between heat received BE and heat rejected DF, all these vertical heights being read on the energy scale. On this diagram the line of absolute zero is also drawn, in order to give an idea of the influence of absolute temperature upon efficiency.

Extension of the two V-curves BC and AD to the V-scale shows the constant specific volume of heating and of cooling. Drawing P-curves from C and D to the P-scale determines the pressures at those points of the cycle. In the same way, a pair of curves from any point on AB or CD will give v and p coördinates for a point on the adiabatic curve of a pressure-volume diagram.

For use with any other than exceptionally accurate and complete tests of combustion engines this graphical method of the single

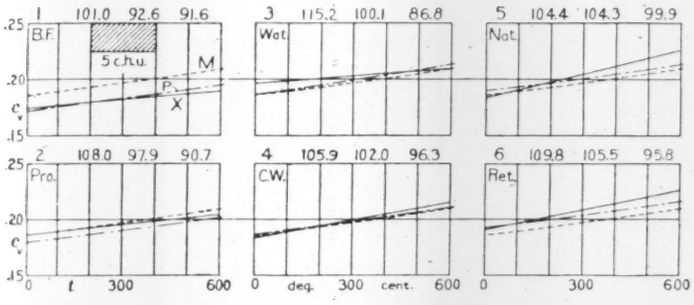

FIG. 6 COMPARISON OF SPECIFIC HEATS c_v OF MIXTURE AND PRODUCTS

chart is sufficiently approximate for practical purposes, in spite of rather wide variation of the properties which are averaged in order to supply its working data. The important influences causing theoretical efficiency to depart from values thus found are: specific heat of both precombustion mixture and products of combustion (Figs. 3 and 6), specific volume (columns 7 and 8 of Table 3), and the change of volume during combustion as expressed by the ratio in column 9 of Table 3.

The influences just named, for the six gaseous fuels of Table 1, are made graphic in Fig. 6, which is a group of individual comparison diagrams. In each case line X (full-line) for the mixture is from Fig. 2 and represents the particular case of Table 4. Curve P (dot-and-dash) for the products is from Fig. 3 and Table 5. Curve M (dotted and the same in all diagrams) represents the mean products, Equation [11]. In general, lines P and X keep fairly close together, showing that the change in specific heat due to the passage through combustion is comparatively small—at least so far as concerns the low range where the specific heat of the mixture is known and necessary. Volumetric information also appears in Fig. 6, each diagram carrying its three ratios from Table 3. Expressed as percentages, these ratios are the volume relative to air of mixture and of products, and the ratio of products to mixture.

Beginning with the second preceding paragraph the paper has been radically revised from the form in which it was first presented. The purposes of this change are to make the presentation clearer and to eliminate an error due to not taking immediately into account the change of specific volume during combustion. The form and effect of that error are indicated later.

COMPUTATIONAL FORM OF THE GRAPHICAL METHOD

In order to make an accurate numerical determination corresponding to Fig. 5 it is necessary to have a computed table of the quantities given by Equations [13] to [16], whether with the coefficients belonging to the average mixture or with those belonging to a particular gas composition. Table 6 is from those equations as written, and has been plotted in the two curves and the u- and i-scales of Fig. 4.

The numerical method of getting cycle dimensions puts special emphasis upon the dimensions of the adiabatic compression and expansion; its first step is to relate the rise or fall of temperature to the volume ratio r_v or the pressure ratio r_p, as illustrated by Fig. 7. In terms of compression AB, ratio $r_v = v_a/v_b$, $r_p = p_b/p_a$, and $r_T = T_b/T_a$. Always these are to be taken in the direction that makes them greater than 1, not fractions, so that operations upon their logarithms will involve only positive numbers. In terms of an expansion from state 1 to state 2,

$$\frac{p_1v_1}{p_2v_2} = \frac{T_1}{T_2} \quad \text{or} \quad \frac{r_p}{r_v} = r_T \quad \text{or} \quad r_p = r_v r_T \ldots\ldots [21]$$

which shows the general relation among these ratios.

Graphical construction for adiabatic compression AB, Fig. 7, begins with curve HA of a given initial pressure, or curve JA of a given initial volume, on either of which point A is located by the height of initial temperature t_a. From this first position the curve is shifted to the left, to a higher pressure along KB or a smaller volume along LB. Entropy distance AE or AF is found by means of Equation [20]. For the coefficient $2.3026AB$ or δ in that equation, since $AB = 1.986/m$, the general value is

$$\delta = 4.573 \div m \ldots\ldots [22]$$

where m is the mean molecular weight of the gas mixture.

The scheme of calculation by means of Table 6 has the effect of referring the whole determination to a single curve, which is KB or LB for the adiabatic compression. With the temperature rise AB along either curve is associated the entropy change EA or FA belonging to an isothermal operation of the same pressure ratio r_p or volume ratio r_v. This is called ϕ_{ab} and distinguished as $EA = \phi_{pab} = \delta \log r_p$ or $FA = \phi v_{ab} = \delta \log r_v$. The procedure, then, is to read ϕ_{pa} or ϕ_{va} in Table 6 at temperature t_a, add ϕ_{pab} or ϕ_{vab} to get ϕ_{pb} or ϕv_b, and read (interpolate) t_b. Taking energy u_b at t_b and subtracting u_a at t_a, the difference is equivalent to the outer work of compression.

As in Fig. 5, add heat of combustion q and get u_c. At the value

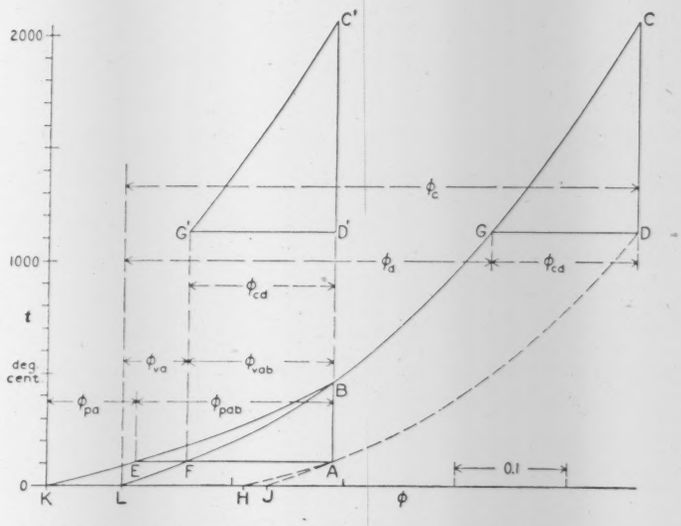

FIG. 7 GRAPHICAL OUTLINE OF CALCULATION OF OTTO CYCLE

of u_c interpolate in Table 6 for t_c and ϕ_c; the distinguishing subscript v may now be omitted, since only volume ratio r_v is determinative of the adiabatic expansion CD, physically, graphically, and computationally. From ϕ_c subtract $\phi_{cd} = \phi v_{ab}$, getting ϕ_d. Inter-

TABLE 6 ENTROPY AND ENERGY VALUES FOR THE MEAN PRODUCTS, CENTIGRADE UNITS

t deg. cent.	ϕ_v	ϕ_p	u c.h.u.	i	t deg. cent.	ϕ_v	ϕ_p	u c.h.u.	
50	0.0313	0.0429	9.3	12.7	1100	0.3261	0.4368	228.6	304.1
100	0.0584	0.0798	18.7	25.6	1200	0.3426	0.4582	252.1	334.4
150	0.0823	0.1123	28.2	38.6	1300	0.3582	0.4785	276.2	365.3
200	0.1038	0.1415	37.8	51.6	1400	0.3734	0.4979	300.7	396.7
250	0.1233	0.1679	47.5	64.7	1500	0.3880	0.5164	325.8	428.7
300	0.1412	0.1920	57.3	77.9	1600	0.4021	0.5342	351.4	461.1
350	0.1577	0.2143	67.2	91.2	1700	0.4158	0.5514	377.5	494.1
400	0.1731	0.2350	77.2	104.7	1800	0.4289	0.5680	404.2	527.7
450	0.1876	0.2544	87.3	118.2	1900	0.4417	0.5841	431.5	561.9
500	0.2013	0.2726	97.5	131.8	2000	0.4543	0.5997	459.6	596.7
550	0.2143	0.2898	107.8	145.5	2100	0.4666	0.6149	488.0	632.0
600	0.2265	0.3062	118.2	159.4	2200	0.4786	0.6298	517.1	668.0
650	0.2382	0.3218	128.7	173.4	2300	0.4904	0.6443	546.8	704.6
700	0.2495	0.3366	139.4	187.4	2400	0.5020	0.6585	577.2	741.9
750	0.2603	0.3508	150.1	201.6	2500	0.5134	0.6724	608.2	779.8
800	0.2705	0.3644	161.0	215.9	2600	0.5246	0.6861	639.9	818.3
850	0.2804	0.3775	171.9	230.3	2700	0.5357	0.6995	672.3	857.5
900	0.2901	0.3902	183.0	244.8	2800	0.5466	0.7127	705.4	897.5
950	0.2995	0.4024	194.2	259.4	2900	0.5574	0.7257	739.1	938.1
1000	0.3086	0.4142	205.6	274.2	3000	0.5681	0.7385	773.6	979.4

polate for t_d and u_d; then $u_c - u_d = u_{cd}$ is the work of adiabatic expansion. Finally, the net work in heat units is

$$Aw = u_{cd} - u_{ab} \quad [23]$$

and the ideal thermal efficiency of the cycle is

$$e = Aw/q \quad [24]$$

A final showing of the essential numerical relationship is made in Fig. 7 by moving expansion diagram CDG over to $C'D'G'$, with $C'D'$ above AB; this makes clear the governing influence of compression entropy ϕ_{vab}. The computations for Fig. 7 are given as a part of the example under Fig. 8, in the second following paragraph.

Computation of the cycle with a particular or actual gas mixture, taking into account all variations in ideal action, is illustrated by Fig. 8. For this there must be computed a table of ϕ_v, ϕ_p, and u

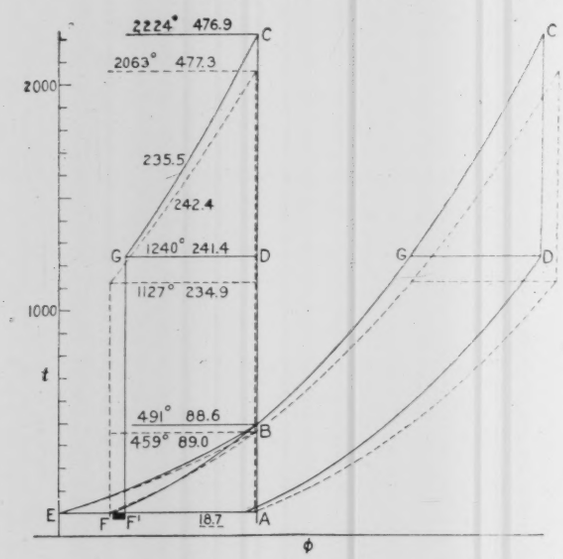

FIG. 8 THE CYCLE WITH A PARTICULAR GAS MIXTURE IN ITS VARIATIONS

for the combustible mixture, and one of ϕ_v and u for the products, as combined in Table 7; and the two values of δ or $2.3026AB$, with their ratio, are required. Having the pressure ratio r_p as a primary datum, the procedure is as follows:

Compute ϕ_{pab} or EA, add it to ϕ_a and get ϕ_{pb};
At ϕ_{po} read t_b and u_b for the mixture;
From the table read ϕ_{vab} or FA, then by the ratio δ_m/δ_p or v_m/v_p convert to $\phi_{vab} = F'A$ for the products; this can also be done by finding r_v by means of Equation [21], then applying Equation [20];
Having ϕ_{vab} for the products, this is also ϕ_{cd}, and the dimensions of expansion CD are found just as under Fig. 7;
In Fig. 8 the dotted outline, reproduced from Fig. 7, shows the same cycle (same in governing data) with mean products properties according to Table 6.

(*A*) For the example of Figs. 7 and 8 the general data are: $t_a = 100$ deg. cent.; $p_a = 13.5$ and $p_b = 180$ lb. abs.; $r_p = 13.33$, $\log r_p = 1.1249$; gas 1 of Tables 1 and 3, with $AB = 0.0692$ and $\delta = 0.1593$ for the mixture and ratio $v_m/v_p = 1.091$; heat of combustion $q = 388.3$ c.h.u. In condensed outline, with first column for the actual composition (Fig. 8) and second column for the mean products (Fig. 7), the computation is as follows:

			Gas 1	Mean
1	Compression entropy	$\delta_m \log r_p = EA = \phi_{pab}$	0.1792	0.1777
2	Initial entropy	ϕ_{pa}	0.0764	0.0798
3	At end of compression	ϕ_{pb}	0.2556	0.2575
4	From Table 7 or Table 6	t_b	490.5	458.5
5		u_b	88.6	89.0
6		u_a	17.6	18.7
7	Work of compression	u_{ab}	71.0	70.3
8	Temperature ratio	$T_b \div T_a = r_T$	2.047	1.961
9	Volume ratio	$r_p \div r_T = r_v$	6.520	6.799
10	By Equation [22]	$\delta_p \log r_v = \phi_{vab}$	0.1297	0.1316
11	From Table 7 or Table 6	ϕ_{vb}	0.1845	0.1899
12		ϕ_{va}	0.0548	0.0584
13	Entropy FA for mixture	ϕ_{vab}	0.1297	0.1316
14	Ratio of specific volumes	$r_m \div r_p$	1.091	1.000
15	Entropy $F'A$ for products	$\phi_{vab} = \phi_{cd}$	0.1189	0.1317
16	Heat of combustion	q	388.3	388.3
17	Add to u_b, line 5, get	u_c	476.9	477.3
18	From Table 7 or Table 6	t_c	2224	2063
19		ϕ_c	0.4417	0.4621
20	Deduct ϕ_{cd}, line 15, get	ϕ_d	0.3228	0.3305
21	From Table 7 or Table 6	t_d	1240	1127
22		u_d	241.4	234.9
23	Deduct from u_c, line 17, get	u_{cd}	235.5	242.4
24	From line 7, corrected for mean gas	u_{ab}	71.0	76.7
25	Net work or ideal output	Aw	164.5	165.7
26	Divide by input q, get efficiency	e	0.424	0.426
27	Discrepancy, in output Aw, per cent.			+0.7

Here temperatures and energy quantities are in centigrade units.

The preceding schedule merely carries out the process just described, and needs but a few comments. Lines 8–10 find temperature and volume ratios, useful for comparison with results from other data, and get volume-ratio entropy ϕ_{vab}; then lines 11–13 make a parallel determination of the last quantity: agreement of the two values is a useful check on computation. Between lines 13 and 15 lies the adjustment emphasized by the shaded block FF' in Fig. 8: 0.1297 from line 13 is divided by 1.091, to get effect of change in AB or δ. The heat of combustion q, line 16, is the full value belonging to the pure mixture, not considering the dilution by waste gases retained in the clearance space. Lines 17 to 23 carry out the general scheme in straightforward way, and lines 25–26 complete it and compare results.

The correction of u_{ab} in line 24 of the calculation with mean gas properties calls for special explanation. In this case the specific volume of the combustible mixture is 1.091 times that of the products of combustion. Then, if other conditions were equal, the compression of the mixture through a given ratio r_p would require 1.091 times as much work as the same compression of the products. To apply this idea to the approximate calculation with mean gas, take $u_{ab} = 70.3$ from line 7, multiply it by 1.091 and get the adjusted value 76.7 for line 24. Of course other things are not quite equal and the "correction" is only a rough one. It is finally justified only by its effect, which is not always so good as here.

This general scheme for finding output places all emphasis upon

TABLE 7 QUANTITIES FOR BLAST FURNACE GAS NO. 6 CENTIGRADE UNITS

Mixture				Products					
t	ϕ_v	ϕ_p	u	t	ϕ_v	u	t	ϕ_v	u
100	0.0548	0.0764	17.6	900	0.2690	169.7	2000	0.4179	419.9
150	0.0771	0.1074	26.5	1000	0.2860	190.4	2100	0.4287	445.0
200	0.0971	0.1351	35.4	1100	0.3020	211.6	2200	0.4392	470.6
250	0.1152	0.1602	44.4	1200	0.3171	233.2	2300	0.4495	496.6
300	0.1318	0.1831	53.5	1300	0.3315	255.1	2400	0.4596	523.0
350	0.1470	0.2041	62.6	1400	0.3453	277.5	2500	0.4695	549.8
400	0.1612	0.2236	71.8	1500	0.3585	300.3	2600	0.4791	577.0
450	0.1745	0.2419	81.0	1600	0.3712	323.4	2700	0.4885	604.5
500	0.1869	0.2589	90.3	1700	0.3835	346.9	2800	0.4978	632.5
550	0.1987	0.2750	99.7	1800	0.3953	370.8	2900	0.5069	661.0
600	0.2098	0.2902	109.2	1900	0.4068	395.1	3000	0.5158	689.9

the two work quantities in compression and expansion, and upon their numerical difference or algebraic sum. An alternative method would be to find the heat rejected during the cooling operation DA and deduct it from received heat q. The first determination has the advantage that it goes around all questions such as are suggested by the failure of the cycle to close at A in Fig. 8.

COMPARISONS OF RESULTS

The things thus far accomplished in this discussion are:

(*a*) Development of the graphical method of Figs. 4 and 5, effective chiefly for the approximate determination by use of the average gas mixture; unless a great deal of routine work is to be done the plotting of a chart like Fig. 4 or Fig. 9 is rather a large task.

(*b*) The establishment of numerical methods which, even when they include the preliminary computation of Table 7, are much more convenient than the direct use of Equations [17] to [19].

(*c*) The beginning of a comparison of the correct and approximate determinations, in one example, and the development of a major correction for the approximate result.

The next step is to try out the approximate as against the close method, using for this purpose eight of the ten examples in Tables 1

to 5. The ones omitted are No. 7, heavy oil, and No. 10, methyl alcohol. By means of condensed tabulations results will be compared, first as found exactly by the form in paragraph (*A*) above, then with a start from the same volume ratio r_v and from the same temperature range t_a-t_b.

In Table 8 the first exhibit is a comparison of energy u at 500 deg. cent., given first for the precombustion mixture and then for the products of combustion; in effect this makes numerical the comparisons in Fig. 6 and extends them to examples 8 and 9. Next,

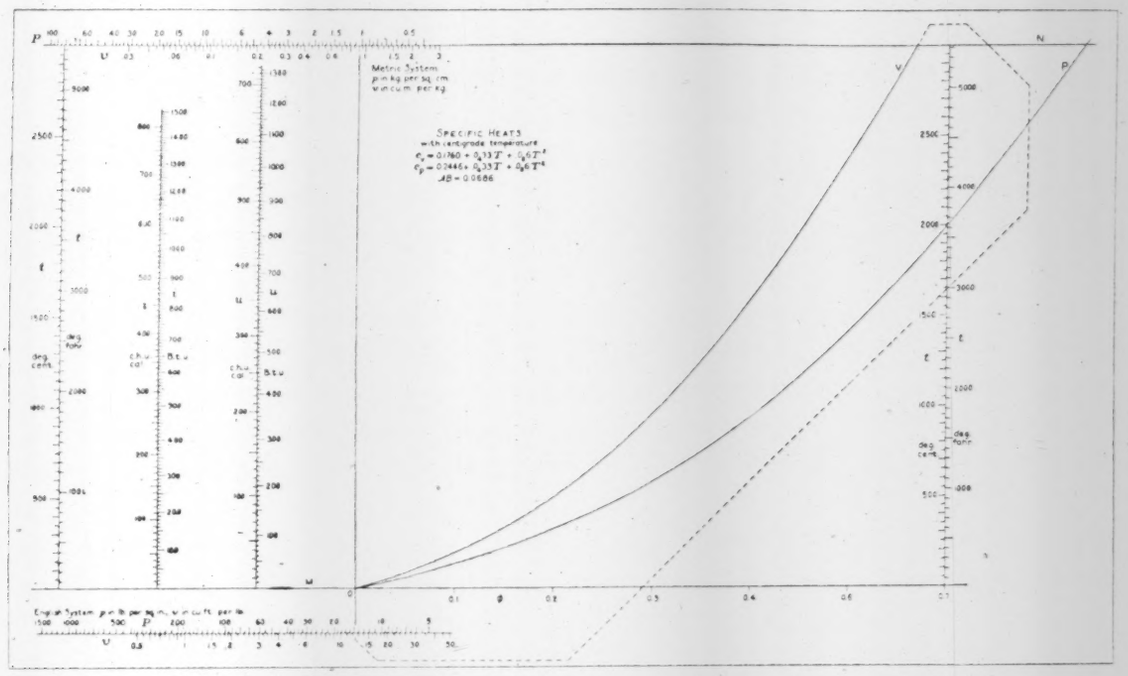

FIG. 9 CHART FOR FINDING IDEAL OUTPUT OF GAS-ENGINE CYCLES
(See Par. (*B*) and footnote.)

TABLE 8 COMPARISON OF GAS-MIXTURE PROPERTIES

No.	1	2	3	4	5	6	8	9	Mean
Fuel	B.F.	Prod.	Water	C.W.	Nat.	Ret.	Gaso.	Alco.	
500 u_m	90.3	95.0	100.6	98.0	100.3	102.5	98.7	96.7	97.5
500 u_p	90.5	94.3	98.5	98.0	99.5	101.0	96.2	96.4	97.5
1000 u_p	190.4	198.4	208.7	206.5	209.6	213.0	202.1	202.4	205.6
2000 u_p	419.9	440.3	468.9	461.9	469.6	478.8	449.8	450.4	459.5
3000 u_p	689.9	733.6	789.2	778.5	795.4	813.8	755.7	759.2	773.6
δ_m	0.1593	0.1679	0.1819	0.1672	0.1649	0.1738	0.1524	0.1545	0.1580
v_m/v_p	1.091	1.087	1.152	1.038	1.001	1.044	0.954	0.958	1.000

by values of u at 1000, 2000, and 3000 deg., the specific heat of each products mixture is compared with that of the average products in the last column. Note how gases 1, 2, 8, and 9 lie below the mean in specific heat, while 4, 3, 5, and 6 run increasingly above it. Finally, the volumetric properties immediately used in the calculation of Par. (*A*) are given in the last two lines; for a fuller showing turn back to Table 3, in its last five columns.

Table 9 gives selected items from computations exactly like the one in Par. (*A*), starting with the same data p_a = 13.5 and t_a = 100; only in Example 2 do the p_b and r_p values not quite correspond with the 13.5 for p_a. The compression pressure p_b is a typical value for each kind of fuel. In each pair of lines after the first two, the first one gives values for the actual gas, and the second for the mean gas, so that the two lines represent the two columns of the calculation in Par. (*A*). Touching upon only a few salient points, comment may be made as follows:

Lines 3 and 6: Larger specific volume, shown by δ_m in all but cases 8 and 9, means more work u_{ab} of compression, tending to raise t_b; lower specific heat tends to make t_b higher, higher specific heat to make it lower; except in cases 8 and 9 these influences so work together as to make t_b higher for the actual mixture than as computed for the mean products.

Lines 4 and 5: These comparisons are interjected in order to suggest the degree of difference that may result from change to equal r_v or equal r_T as primary datum.

Line 9: Here specific heat has controlling influence, so that in cases 3 to 6, temperature t_c is less for the actual gas than for the mean gas.

Line 12: The first u_{ab} is repeated from line 6, the second is adjusted as explained earlier.

Lines 13 and 15: Here is the real test of the approximate method; as in line 27, Par. (*A*), the discrepancy is expressed as a percentage of the actual output. Example 3 is anomalous, but fortunately that gas is not an engine fuel; the other gas-fuel examples show a discrepancy ranging up to 1 per cent, while the liquid fuels are determined about 3 per cent low by the approximate method.

Line 14: This efficiency is for the actual gas, line 13*a*.

The next condition to be considered is that in which volume ratio r_v is the same in both calculations, exact and approximate. As a datum this is simpler than r_p, because r_v for the mixture need only be adjusted by v_m/v_p in order to become r_v for the products. For Table 10 the general data p_a, t_a, and q are continued, and r_v is taken at about the mean of the two values in Table 9. On the whole, for the gas fuels agreement is not quite so good as in the preceding set of calculations; in cases 4 and 6 it would really be a little better without the adjustment of u_{ab} by v_m/v_p. In the general run there is an average discrepancy of about —1.7 per cent, say, 1 in 60.

In Table 9, line 5, it is seen that with the same r_p there is a comparatively small difference between the two r_v's for a given gas, by the exact and approximate methods. In Table 11, starting

TABLE 9 OUTLINED COMPUTATIONS OF THE EIGHT EXAMPLES, EACH FOR ACTUAL AND FOR MEAN COMPOSITION

No.		1	2	3	4	5	6	8	9
1	p_b	180	150	90	100	135	135	100	162
2	r_p	13.33	11	6.67	7.41	10	10	7.41	12
3*a*	t_b	490.5	452.2	366.5	370.0	411.8	421.4	345.3	432.2
3*b*		458.8	424.0	340.3	357.0	407.2	407.2	357.0	439.4
4*a*	rT	2.047	1.944	1.715	1.722	1.836	1.862	1.658	1.897
4*b*		1.961	1.869	1.644	1.689	1.824	1.824	1.689	1.910
5*a*	r_v	6.520	5.658	3.888	4.301	5.447	5.371	4.469	6.326
5*b*		6.819	5.887	4.055	4.386	5.484	5.484	4.386	6.283
6*a*	u_{ab}	71.0	67.2	53.5	52.8	62.7	66.1	48.2	64.4
6*b*		70.3	63.4	46.6	49.9	60.0	60.0	49.9	66.5
7	q	388.3	470	660	625	575	610	550	435
8*a*	u_c	476.9	555.6	733.2	696.4	656.3	695.5	617.1	518.1
8*b*		477.3	552.1	725.3	693.6	653.7	688.7	618.6	520.2
9*a*	t_c	2224	2416	2839	2760	2600	2672	2573	2239
9*b*		2063	2317	2859	2764	2642	2750	2533	2210
10*a*	t_d	1240	1450	2042	1892	1636	1717	1695	1253
10*b*		1127	1383	2022	1897	1677	1758	1699	1271
11*a*	u_{cd}	235.5	254.1	251.9	264.7	288.1	298.2	248.3	257.7
11*b*		242.4	255.6	259.7	262.9	282.3	295.6	241.2	251.1
12*a*	u_{ab}	71.0	67.2	53.5	52.8	62.7	66.1	48.2	64.4
12*b*		76.7	68.9	53.8	51.8	60.1	62.6	47.6	63.7
13*a*	Aw	164.5	186.9	198.4	211.9	225.4	232.1	200.1	193.3
13*b*		165.7	186.7	205.9	211.1	222.2	233.0	193.6	187.4
14	e	0.424	0.397	0.301	0.339	0.392	0.380	0.364	0.444
15	disc.	+0.7	—0.1	+3.8	—0.4	—1.4	+0.4	—3.2	—3.1

with a given temperature range in each case, there appears a large reaction upon r_p and r_v. The effect of this relationship is shown by the results in lines 4 and 5, which are given without any of the intermediate steps in the calculation. The output values in line 4 are computed without paying any attention to differences in r_v, or with the tacit assumption that ϕ_{cd} is the same for the actual gas, line 4*a*, and for the mean gas, line 4*b*. Really and correctly taking the factor v_m/v_p into account, the actual gas yields the output in line 5*a*, from which line 4*b* is discrepant by as much as 10 per cent in the worst case. Dividing result 4*b* by the ratio v_m/v_p gives the adjusted values in line 5*b*, which are in good accord with 5*a*.

Failure to perceive this effect of specific-volume change was a serious error in a portion of the paper as originally presented,

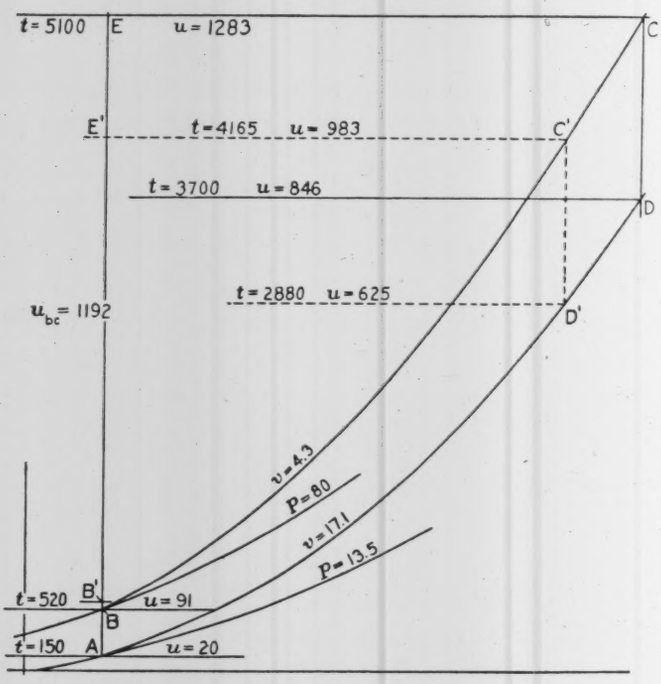

FIG. 10 CONSTRUCTION FOR THE OTTO CYCLE, GAS EXAMPLE NO. 3

although that error had little if any effect upon the final method there presented. In any case, that final method has now been a good deal simplified in its exact form.

TABLE 10 RESULTS FROM EQUAL VOLUME RATIO

No.		1	2	3	4	5	6	8	9
1	r_v	6.5	5.7	4.0	4.3	5.5	5.5	4.4	6.3
2*a*	t_b	489.5	453.8	373.1	370.0	414.1	427.0	344.8	433.0
2*b*		447.3	416.2	337.3	352.6	408.0	408.0	357.8	440.0
3*a*	u_{ab}	70.8	67.5	56.7	52.7	63.2	67.3	47.6	64.6
3*b*		68.0	61.8	46.1	49.1	60.1	60.1	50.1	66.6
4*a*	u_c	476.7	555.9	736.4	696.3	656.8	696.7	616.5	518.3
4*b*		475.0	550.0	724.8	692.8	653.8	688.8	618.8	520.3
5*a*	t_c	2223	2417	2848	2760	2602	2675	2571	2210
5*b*		2055	2312	2857	2762	2643	2750	2533	2211
6*a*	t_d	1235	1448	2037	1893	1632	1708	1702	1256
6*b*		1140	1393	2028	1906	1675	1764	1696	1270
7*a*	u_{cd}	235.9	255.0	259.5	264.4	289.6	301.9	246.3	257.4
7*b*		237.0	250.3	257.2	259.6	283.6	294.0	242.6	251.3
8*a*	u_{ab}	70.8	67.5	56.7	52.7	63.2	67.3	47.6	64.6
8*b*		74.2	67.3	53.1	51.0	60.2	62.7	47.8	63.8
9*a*	Aw	165.1	187.5	202.8	211.7	226.4	234.6	198.7	192.8
9*b*		162.8	183.0	204.1	208.6	223.4	231.3	194.8	187.5
10	e	0.425	0.399	0.307	0.339	0.394	0.384	0.362	0.443
11	disc.	—1.4	—2.4	+0.6	—1.5	—1.3	—1.4	—2.0	—2.8

A good agreement between the approximate and the exact solution, as in most cases of Tables 9 and 10, means generally that the influences complicating real performance have an interplay that makes them more or less balance each other. In Table 11 we see how they may so interact as to cause a big discrepancy. The approximate solution is much more serviceable for finding output than for cycle dimensions.

TABLE 11 THE EFFECT OF USING EQUAL TEMPERATURE RANGE IN THE ADIABATIC COMPRESSION

No.		1	2	3	4	5	6	8	9
1	t_b	480	450	350	370	410	420	350	480
2*a*	r_p	12.68	10.89	6.09	7.41	9.90	9.92	7.63	11.70
2*b*		14.93	12.74	7.10	8.01	10.16	10.76	7.10	11.40
3*a*	r_v	6.28	5.62	3.65	4.30	5.41	5.35	4.57	6.20
3*b*		7.41	6.57	4.25	4.68	5.55	5.79	4.25	6.04
4*a*	Aw	177.4	203.4	229.7	219.9	222.7	242.2	192.9	182.4
4*b*		177.5	201.4	219.0	221.6	223.1	238.4	188.4	181.5
5*a*	Aw	161.6	186.4	198.1	211.9	222.1	231.4	201.9	190.8
5*b*		162.7	185.3	189.9	213.6	222.8	228.3	197.5	189.5

METHOD FOR THE OTTO CYCLE

(*B*) The working chart for approximate determinations is pictured in Fig. 9, but in working form is nearly four times that size.[9] It is like Fig. 4 in general terms, with all of its curves and scales based on the properties of the mean products of combustion, as stated earlier and as computed (partly) in Table 6. The scales are so disposed that any problem can be constructed upon a separate sheet of paper, laid down from above across the space between the u scale and the right-hand scale of t, the templet guide lines M and N being drawn across this problem sheet. Of course the chart is to be laid down on the drawing board, with use of the T-square for horizontals of constant t, u, or i. The complete form of one of the heavy-paper templets, that for curve V, is shown in dotted outline. Having the metric P- and V-scales laid out along the upper guide line (but to be projected to line M as needed), this chart will give results in any one of the three measurement systems named at the beginning of the paper.

The graphical example in Fig. 10 is worked in English-fahrenheit units and starts with the data $p_a = 13.5$, $p_b = 80$, and $t_a = 150$. After drawing the two p-constant curves and locating compression AB, the two v-constant curves AD and BC are drawn. Of course, the volume values lettered on these curves, as read from the v-scale of the chart, are only very roughly approximate for the actual mixture, with its specific-volume ratio 1.152. Adiabatic expansion CD is located just as in Fig. 5, by adding q or u_{bc} to u_b to get u_c.

In order to give an idea of probable accuracy by graphical determination, the following tabulation compares the results on Fig. 10 with those computed by means of a fahrenheit-unit Table 6:

	u_a	t_b	u_b	u_c	t_c	t_d	u_d	u_{cd}	u_{ab}	Aw
Const.	20	520	91	1283	5100	3700	846	437	82	355
Calc.	22.1	516	92.2	1284.2	5115	3715	850.5	433.7	81.7	352

As might be expected, it appears that the construction can easily accumulate an error of the same order of magnitude as the discrepancies in Tables 9 and 10.

The increase of compression range AB to AB' represents the "correction" for a larger specific volume of mixture, as described in Par. (*A*) et seq. The dotted expansion line $C'D'$ shows the determination when dilution of mixture by waste gases (in clearance space) is taken into account. As a fair approximation, assuming exact geometrical scavenging and that the influences of pressure difference, temperature difference, and density difference balance each other, the weights of fresh mixture and retained gases are taken to be proportional to displacement volume and clearance volume. Using the volumes on Fig. 10, the ratio of these volumes is here $4.3 \div 17.1 = 0.25$. Its chief influence is to diminish the effective heating value, of the pound of actual mixture in the cylinder, from 1192 to $1192 \times 0.75 = 892$ B.t.u. This new value determines the height of line $E'C'$ and the location of $C'D'$. A second effect is to increase slightly the specific heat and the relative volume of the mixture, in the direction of making the mixture more like the products of combustion in these properties.

METHOD FOR THE DIESEL CYCLE

In discussing this cycle a purely constant-pressure combustion will be assumed, at the pressure p_b due to the adiabatic compression of the charge of air. That charge will be the full amount of air entering into the cycle, which means practically that the engine has solid injection, not air injection, of the fuel oil. Both physically and computationally, conditions are more complex than in the Otto cycle, because the compressed charge differs more widely from the products in specific heat and has a different mass, and because the ratio of adiabatic expansion differs from that of compression by an amount that is dependent upon several variable conditions of working.

For air, evaluating Equation [6*b*] according to the particular constants given, with centigrade temperature,

$$c_v = 0.1708 + 0.0_7107T^2, \qquad c_p = c_v + 0.0686 \ldots\ldots [25]$$

[9] Copies of the full-size chart separately printed, may be obtained for a nominal charge, by addressing the Secretary of the Society.—EDITOR.

Assuming as fuel a pure hydrocarbon with the weight composition hydrogen 0.15 and carbon 0.85, the specific heats of the products, with 50 per cent of excess air, are

$$c_v = 0.1740 + 0.0_4239T + 0.0_8382T^2, \quad c_p = c_v + 0.0691 \ldots [26]$$

In the gas equation $pv = bT$, with p in pounds per square inch, T in degrees centigrade, and for v in cubic feet per pound, the values of constant b are,

For air and for mean products, $b = 0.6664$ ..... [27a]

For actual products, $b = 0.6713$ ..... [27b]

A comparison of values of energy u, like that in Table 8, gives the following:

	t = 500	1000	1500	2000	2500
Air	87.0	178.0	275.9	383.3	502.9
Actual products	93.8	195.0	304.7	423.8	553.1
Mean products	97.5	205.6	325.8	459.6	608.2

Tables like Nos. 6 and 7, for the two gas compositions of this example, are combined in Table 12.

A double example will now be given, starting first with the same pressure range, then with the same volume range, in adiabatic compression AB. With the fuel composition stated in the second preceding paragraph the weight ratio is 22.02 lb. of air to 1 lb. of oil, or the air compressed for 1 lb. of total medium is 0.9566 lb.; this number is the mass ratio r_m. General data are, p_a = 13.5 lb. abs., t_a = 75 deg. cent., p_b = 600 lb. abs. in case (1), r_{vab} = 16 in case (2). In either case the first column is for actual compositions, Table 12, the second is an approximate calculation by means of Table 6. Instead of being made continuous, as in Par. (*A*), this outlined calculation is broken into sections, with explanations as needed.

In the adiabatic compression, with r_p = 600 ÷ 13.5 = 44.44, the entropy ϕ_{pab} is 0.1580 × 1.6478 = 0.2604 for both columns of case (1); with r_v = 16 the similar common value is

$$\phi_{vab} = 0.1580 \times 1.2041 = 0.1903$$

for case (2). Working at first with the full pound of air (as to entropy values), the two calculations are:

Case (1)		Actual	Mean
1 Initial entropy	ϕ_{pa}	0.0577	0.0614
2 Entropy of pressure range	ϕ_{pab}	0.2604	0.2604
3 Entropy at B	ϕ_{pb}	0.3181	0.3218
4 Temperature at B	t_b	737.5	650.0
5 Temperature ratio $T_b \div T_a$	r_T	2.904	2.652
6 Volume ratio, by Equation [21]	r_v	15.31	16.76

Case (2)		Actual	Mean
1 Initial entropy	ϕ_{va}	0.0414	0.0449
2 Entropy of volume range	ϕ_{vab}	0.1903	0.1903
3 Entropy at B	ϕ_{vb}	0.2317	0.2352
4 Temperature at B	t_b	755.6	637.2
5 Temperature ratio	r_T	2.973	2.616
6 Pressure ratio	r_p	47.57	41.85
7 Pressure at B	p_b	642.2	564.9

In case (1) the difference in specific heat causes a large difference in temperature t_b, with incidental inequality of volume ratio; then equalizing the volume ratio makes the temperature difference yet larger in case (2).

From here on the two calculations follow the same schedule and can be carried forward in parallel columns. All the energy quantities belonging to the adiabatic compression (next to be given) are read from the table for the pound of medium, then diminished by the factor r_m or 0.9566; practically, from each is deducted the fraction 0.0434 of itself. Besides compression work u_{ab} we now need enthalpy or "total heat" i_b, also the "outer work" $APv = i_b - u_b = o_b$; the last is for use in finding the work of constant-pressure expansion BC.

		Case (1)		Case (2)	
Adiabatic compression		Actual	Mean	Actual	Mean
8 Enthalpy at B	i_b	172.4	165.9	176.8	162.4
9 Energy at B	u_b	124.0	123.1	127.1	120.5
10 Energy at A	u_a	12.3	13.4	12.3	13.4
11 Compression work	u_{ab}	111.7	109.7	114.8	107.1
12 Outer work at B	o_b	48.4	42.8	49.7	41.9

As concerns physical data, the most uncertain point in the whole determination is the change of enthalpy, at constant pressure p_b and temperature t_b, from r_m lb. of air to 1 lb. of air plus vaporized oil. Lacking definite knowledge of the thermal properties of average Diesel-engine oil, the approximation will here be made of simply adding the heat of combustion q (itself no more than an average value) to enthalpy i_b of the compressed charge (line 8 above) and taking this to be i_c at the end of combustion. Per pound of total medium or per 0.0434 lb. of oil, the heating value here used is 435 c.h.u. Continuing the calculation,

		(1)		(2)	
Constant-pressure combustion					
13 Enthalpy at C	$i_b + q = i_c$	607.4	600.9	611.8	597.4
14 Temperature at C	t_c	2142	2012	2156	2002
15 Energy at C	u_c	459.2	462.9	462.6	460.2
16 Outer work at C	o_c	148.2	138.0	149.2	137.2
17 Work of expansion	$o_c - o_b = o_{bc}$	99.8	95.2	99.5	95.3

The increase of volume from B to C is due to increase of mass, change of specific volume through combustion (here very small),

TABLE 12 QUANTITIES FOR DIESEL-ENGINE MEDIA CENTIGRADE UNITS

Air					Products				
t	ϕ_v	ϕ_p	u	i	t	ϕ_v	ϕ_p	u	i
50	0.0288	0.0403	8.6	12.0	1000	0.2948	0.4011	195.1	264.2
100	0.0537	0.0751	17.2	24.1	1100	0.3109	0.4224	216.3	292.3
150	0.0754	0.1054	25.8	36.1	1200	0.3260	0.4424	237.9	320.8
200	0.0947	0.1324	34.5	48.2	1300	0.3404	0.4613	259.8	349.6
300	0.1280	0.1789	51.8	72.4	1400	0.3542	0.4793	282.1	378.9
400	0.1561	0.2181	69.3	96.8	1500	0.3673	0.4965	304.7	408.4
500	0.1806	0.2520	87.0	121.3	2000	0.4263	0.5727	423.8	562.0
600	0.2022	0.2820	104.8	145.9	2100	0.4371	0.5864	448.7	593.8
700	0.2218	0.3090	122.8	170.8	2200	0.4476	0.5998	474.2	626.2
800	0.2396	0.3333	141.0	195.9	2300	0.4579	0.6128	500.1	659.0
900	0.2560	0.3460	159.4	221.2	2400	0.4679	0.6254	526.4	692.2
1000	0.2713	0.3769	178.1	246.7	2500	0.4777	0.6378	553.1	725.9

and rise of temperature. The two volumes, per pound of total medium, are

$$v_b = \frac{r_m b_m T_b}{p_b}, \qquad v_c = \frac{b_p T_c}{p_c} \ldots [28]$$

and since $p_b = p_c$, the volume ratio is

$$r_{vbc} = \frac{1}{r_m} \frac{b_p}{b_m} \frac{T_c}{T_b} \ldots [29]$$

Here the subscripts m and p on gas constant b refer to mixture and products, as in Table 3. With the values of b as in Equations [27a] and [27b] and of r_m as given at the beginning of the example, the coefficient of the temperature ratio is,

For actual gases, 1.0530
For the mean gas, 1.0455.

Dividing r_{vbc} into total r_v, line 6 in case (1), primary datum in case (2), we get the ratio r_{vcd} of adiabatic expansion CD, from which comes the entropy distance that appears as EF or GD in Fig. 11. Using this scheme, which goes around the impossibility of representing on the T-ϕ diagram the changes in mass and volume that take place through combustion, the calculation is completed as follows:

		(1)		(2)	
Adiabatic expansion, etc.					
18 Temperature ratio	T_c/T_b	2.390	2.476	2.363	2.499
19 Volume ratio, range BC	r_{vbc}	2.517	2.589	2.488	2.613
20 Volume ratio, range CD	r_{vcd}	6.082	6.474	6.431	6.123
21 Entropy change	ϕ_{vcd}	0.1247	0.1282	0.1285	0.1243
22 Entropy at C	ϕ_{vc}	0.4415	0.4557	0.4430	0.4545
23 Entropy at D	ϕ_{vd}	0.3168	0.3275	0.3145	0.3302
24 Temperature at D	t_d	1139	1108	1124	1125
25 Energy at D	u_d	224.7	230.6	221.5	234.5
26 Work of expansion	$u_c - u_d = u_{cd}$	234.5	232.3	241.1	225.7
27 From line 17	o_{ba}	99.8	95.2	99.5	95.3
28 Gross work		334.3	327.5	340.6	321.0
29 Negative work, line 11	u_{ab}	111.7	109.7	114.8	107.1
30 Net work or output	Aw	222.6	217.8	225.8	213.9
31 Efficiency	e	0.512	0.500	0.519	0.491
32 Discrepancy, per cent			—2.4		—5.3

In case (1) the actual output is fairly well represented by the approximate result which depends wholly upon the mean-gas properties in Table 6. In case (2) the degree of approximation is poor.

Here, as with the Otto cycle, a good agreement (when obtained) is due to a balancing of disturbing influences.

The final graphical construction, made by means of the chart pictured in Fig. 9, is shown in Fig. 11, for case (1) of the preceding example. It is simpler in idea than the calculation there made and gives a better balance of disturbing influences and a closer result. All of the graphical work is done as if for a full pound of medium in the cylinder, with the coördinates of point B corresponding to that assumption; and the smaller mass during compression is taken into account only in diminishing work u_{ab} to the value given in line 11 of the preceding tabulation by the use of mass

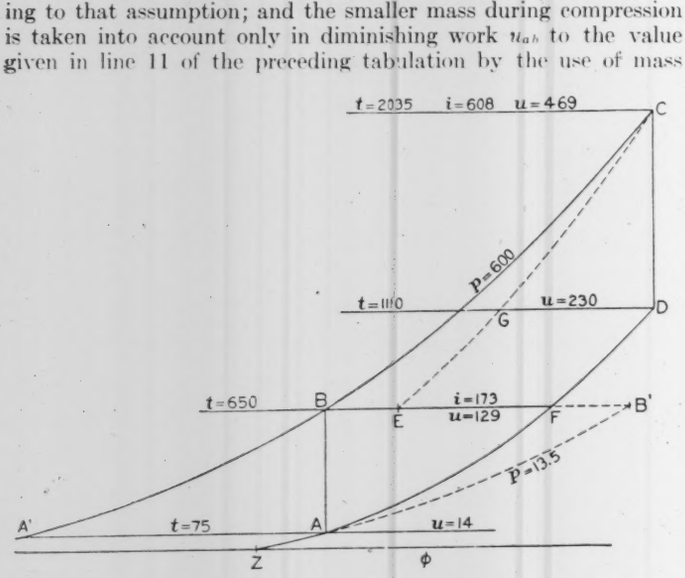

FIG. 11 CONSTRUCTION FOR THE DIESEL CYCLE

factor r_m. In a condensed outline, showing only results, computation of this construction for the two cases in question gives the following:

	Graphical Determination		(1)	(2)
1	One-pound enthalpy at B	i_b	173.4	169.8
2	Add q, get enthalpy at C	i_c	608.4	604.8
3	Temperature at C	t_c	2032	2023
4	Energy at C	u_c	469.0	466.1
5	Temperature at D	t_d	1117	1121
6	Energy at D	u_d	232.6	233.4
7	Constant-pressure work	o_{bc}	94.7	94.9
8	Adiabatic-expansion work	u_{cd}	236.4	232.7
9	Gross out-stroke work		331.1	327.6
10	Compression work, line 11	u_{ab}	109.7	107.1
11	Net work	Aw	221.4	220.5
12	Exact result, line 30		222.6	225.8
13	Discrepancy, per cent		—0.5	—2.4

The working scheme, with p_a and p_b as data, is then to construct Fig. 11 as shown and find output Aw from the scale readings recorded on that diagram. Practically, the construction need not extend to the left of line AB, this compression being equally well determined by "triangle" $AB'B$. Volume curve CE is put in rather to illustrate a step in the computation than because it is needed graphically. With the volume ratio as primary datum the proper procedure is to make the "actual" computation of case (2), then go to the chart with the value of p_b thus found. Computation of output from the readings on Fig. 11 may be slightly shortened, as follows:

$$\begin{aligned} Aw &= (i_c - u_c - i_b + u_b) + (u_c - u_d) - r_m(u_b - u_a) \\ &= (i_c - i_b) - (u_d - u_b) - r_m(u_b - u_a) \quad \ldots\ldots\ldots\ldots [30] \\ &= 435 - 101 - 110 = 224 \end{aligned}$$

This result is a little large, chiefly because point D reads low as compared with its computed height.

Because the best mixture proportion for the Diesel engine is subject to much less variation than in the wider class of Otto-cycle engines, the "exact" properties in Table 12 are very fairly representative for general use. As with the Otto cycle, it is to be understood that the construction in Fig. 11 is claimed to be a good approximation only as to overall output, not as to the details of the cycle and the coördinates of its "points."

Discussion at Oil and Gas Power Session

IDEAL GAS-ENGINE CYCLES

F. O. Ellenwood,[10] wrote that while he agreed with the author regarding the desirability of some short method of making the necessary calculation of cycles involving variable specific heats of the mixture of fuel and air, he was not convinced that the author's scheme was entirely satisfactory.

The ratio of the thermal efficiency of the actual and ideal engines was a true measure of the engine efficiency only when both of these thermal efficiencies had been accurately determined. A variation of 5 per cent in the ideal cycle efficiency might easily change the engine efficiency from 10 to 11 per cent. Thus if an internal-combustion engine had a thermal efficiency of 30 per cent and the corresponding ideal cycle efficiency was 40 per cent, the engine efficiency was 30/40 or 75 per cent. If, however, the cycle efficiency was 35 per cent, the engine efficiency became 30/35 or 85.7 per cent instead of 75 per cent.

Evidently the proposed scheme of using the templets was intended to be much more convenient than using the thermodynamic equations, and it was implied that the accuracy of results would be satisfactory. To Professor Ellenwood it seemed that for commercial engineering applications it would be far simpler and much more accurate to prepare for each fuel used a family of curves that showed the ideal cycle efficiencies for the various ratios of compression and whatever percentages of excess air that might be involved. Such curves probably would give results that were more accurate than the method outlined in the paper, and after they were once calculated their use was far simpler than the proposed scheme. It was true that he preparation of a comprehensive set of curves for general use involved a large amount of labor, but when

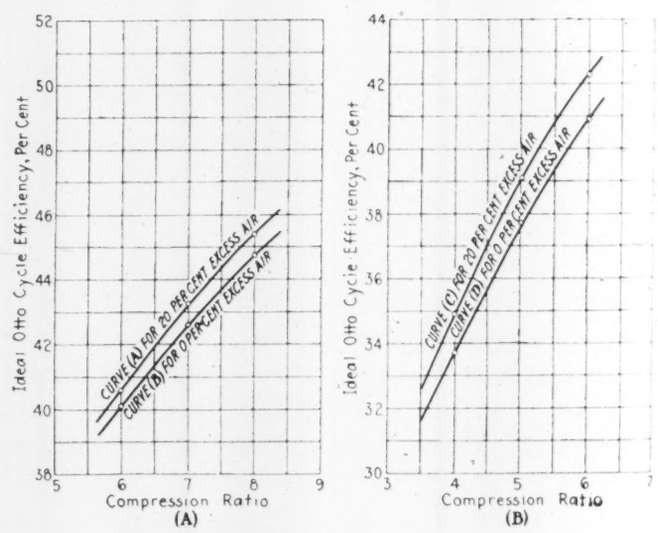

FIG. 12 CURVES OF IDEAL CYCLE EFFICIENCIES FOR OTTO CYCLE
A, results with blast-furnace gas. *B*, results with gasoline.

it was once done correctly, the results were available for indefinite use without further delay.

As evidence of how conveniently the ideal cycle efficiencies might be plotted, curves *A*, *B*, *C*, and *D* in Fig. 12 for the Otto cycle showed the efficiences obtained by the long method of calculation with the variable specific heats of Goodenough and Felbeck, whose values were probably the most reliable of any now available.[11] The compression ratio was one of the vital factors in these efficiencies and it was therefore plotted as one coördinate and might be extended over whatever range was needed. The other factor was the mixture ratio or the percentage of excess air, because it was this ratio that determined the specific heat of the mixture for any specified temperature. In Fig. 12 the curves had been drawn for zero and 20 per cent excess air, but the range could be extended as needed. For the Diesel cycle the percentage of excess air needed

[10] Professor of Heat Power Engineering, Cornell University, Ithaca, N. Y. Mem. A.S.M.E.

[11] Bulletin 139, University of Illinois, 1924.

to cover a much wider range than for the Otto cycle, because at light loads the Diesel engine used a very high ratio of air to fuel.

From Table 10 and the final calculation of the paper, it appeared that the ideal cycle efficiency of the Otto and Diesel engines using different fuels with a small amount of excess air might be calculated by the approximate method of the author to within about 2 per cent of the correct result. For the Diesel under light loads the air-fuel ratio would be widely different from the cases worked out, and the approximate method might therefore be more in error than was shown in the calculation. The error involved might not be objectionable in many applications, and if appreciable time was saved by using this approximate method, it became justified in such cases. However, it seemed to Professor Ellenwood that this approximate method itself would probably be found to involve too much labor to make it generally applicable. Approximate methods were usually successful chiefly because of their extreme simplicity and convenience.

H. A. Everett,[12] wrote that the need for a standard was obvious to all who had done any work in fundamentals. The only standard for comparing internal-combustion-engine performance that we had to date that had had any wide acceptance had been the old "Air Standard" proposed in 1905 by a committee of the Institution of Civil Engineers of Great Britain. At that time they had encountered the same difficulties that we were now meeting. These had been overcome by making five sweeping assumptions: First, the material should have instantaneous action (for example, the combustion should be complete and instantaneous); second, there should be no transfer of heat during the expansion and compression; third, for any given pressure and temperature there was no change in the specific volume of the fluid; fourth, there was no change in the chemical constitution of the fluid; and, fifth, there was no change in specific heat.

For any prospective criterion the first and second assumptions could still be retained, and should be, because they were optimum conditions. The desirability of discarding the third and fourth assumptions, involving, respectively, no change in specific volume and no change of chemical characteristics, was still a debatable point. The author considered it important to take cognizance of both of these effects, and did so, but at the expense of considerable increase in the complexity of the solution. Personally Professor Everett preferred to retain these simplifying assumptions, even though it involved a slight departure from the ideal cycle using actual gas characteristics, because of the enormous simplification of computation. The fifth assumption of the old Air Standard, namely, that relating to the variation of specific heats, was without any question the most erroneous of all, and could not be dropped from consideration.

The suggestion that the author made of using an average gas as typical, or sufficiently typical, to serve as a standard for all conditions, was open to possible pitfalls in conditions arising that might depart, perhaps radically, from the average that he suggested.

It was true that there was an appreciable divergence between the ideal efficiency, considering the working substance as air and as an actual mixture as suggested by the author, but did not the added simplicity resulting from assuming air as the working fluid, provided its variation in specific heat was taken into account, make it a standard preferable to that proposed? It involved no selective judgment and yet gave an ideal efficiency not much divergent from that obtained by considering the actual mixture or the hypothetical mixture proposed by the author.

To illustrate the magnitude of the differences when the ideal efficiency was figured by these different standards the following example had been computed. For a five-to-one compression ratio the efficiency according to the old Air Standard was 47.5 per cent (this with constant specific heat); the efficiency according to the air standard with variable specific heat was 42.6 per cent; and the efficiency taking the actual mixture in detail was 37.1 per cent.[13]

The author, in closing, wrote that as to accuracy, according to the set of representative examples in Tables 9 and 10, the ideal efficiency of the Otto cycle, with best conditions of working (as to air proportions), ranged from 0.30 to 0.45. The chart, carefully handled, would determine such an efficiency within the range from +0.005 to —0.015 with respect to the correctly computed value. This would be true even if no correction was made for the change in specific volume through combustion; and unless the gas fuel was unusually high in H_2 and CO that correction might very well be omitted. Computation of the chart determination, using Table 6 as shown in Par. (*A*), would considerably shorten the range of error, by eliminating graphical inaccuracies, and was not at all a complex operation.

As to complexity, the method of the computed table, as proposed in the paper, was nothing else than the easiest way of applying and satisfying general Equations [17] to [19]. What Fig. 7 really did was to give a physical meaning, in terms of entropy quantities, to the two members of either Equation [17] or [18]. The author had done quite enough work by the direct method—which involved simplification and trial solution—to realize very fully how much easier the new method was.

Wishing to construct such curves as were shown by Professor Ellenwood, the author would proceed as follows, after making the preliminary calculations on the gas mixture:

By means of a chart determination get the approximate temperature height at each of points *B*, *C*, and *D*—unless such knowledge had already been established by previous calculations;

With the help of a special table given in the Appendix, compute a few sets of values in the region of each point;

Compute by the form in Par. (*A*) (Otto cycle) or that at the close of the paper (Diesel cycle).

A Temperature-Entropy Diagram for Air and the Diatomic Gases O_2, N_2, and CO

R. C. H. Heck[14] wrote that problems I and II in Professor Everett's paper illustrated a correct application of the charts. It was questionable, however, whether their application to the determination of the ideal Otto-cycle efficiency was of real value.

The air-standard efficiency was developed upon the assumption that air was the "working medium," and also that it followed the laws of Boyle and Charles at all temperatures, the specific heat of the air being assumed constant. This efficiency was then a function of the clearance volume of the actual engine, expressed as a fraction of the whole cylinder volume, and was very easy to calculate. It was an imaginary quantity, exactly to the same degree as a Carnot engine was imaginary *but it had the virtue of being definite*. The "ideal efficiency," as calculated by Rosecrans and Felbeck (Bulletin 150 of the University of Illinois Experiment Station) quoted in the paper, was based upon careful and complete measurements during the test of a gas engine. This "ideal efficiency" was defined in the Bulletin referred to as follows:

> The ideal thermal efficiency is the ratio of the work obtained from the ideal Otto cycle, computed as shown later, to the work equivalent of the lower heat of combustion of the fuel used. This ideal thermal efficiency is based on the assumption of complete combustion at the time of maximum pressure in the cycle.

"The work obtained from the ideal Otto cycle," as calculated by Rosecrans and Felbeck, might be defined concisely as follows:

The foot-pounds of work shown on an indicator diagram, which *would* be obtained from an engine of the same cylinder proportions as the engine tested, and which was supplied with the same fuel and air, compressed under isentropic adiabatic compression, combustion being completed at the instant of maximum pressure, and the gases expanded under the isentropic adiabatic law; account being taken of the true thermal characteristics of the air-fuel charge, including the burned gases in the clearance space at beginning of suction stroke, depending upon a knowledge of the actual air-fuel ratio and complete volume analysis of the fuel and air; and account also being taken of the true thermal characteristics of the expanding gases, which consisted not only of diatomic gases but also CO_2 and H_2O. The whole calculation, it would be observed, was dependent upon the initial temperature in the cycle, from which the other temperatures must be calculated, assuming the ideal conditions of adiabatic isentropic, and constant-volume changes,

[12] Professor of Thermodynamics, Pennsylvania State College, State College, Pa. Mem. A.S.M.E.

[13] For details of this comparison, see A Temperature-Entropy Diagram for Air and the Diatomic Gases O_2, N_2, and CO, by H. A. Everett, in MECHANICAL ENGINEERING, vol. 48, no. 11a, Nov. (Part 2), 1926, p. 1329.

[14] Professor of Mechanical Engineering, Rutgers University, New Brunswick, N. J. Life Mem. A.S.M.E.

but allowing for the true specific heats of the actual gas constituents. The ideal efficiency thus defined had the following disadvantages:

1 It was exceedingly laborious to calculate
2 It was out of the question, in commercial tests, to obtain such complete experimental data as to make the calculation possible
3 It was *not* the ideal efficiency of the engine, but the combined ideal efficiency of the engine as modified by actual fuel-charge characteristics. If the air-fuel ratio or if the nature of the fuel was changed, the "ideal efficiency" also changed.
4 It was dependent upon a calculated (initial) actual temperature at a point on the compression stroke. This was *not* the temperature under *ideal* conditions, but under *actual* conditions. It was established by an *actual* pressure as measured from an indicator diagram.

The ideal efficiency presented in the paper discussed was that of an Otto cycle using diatomic gases (air), taking account of the variable specific heat of the air, but ignoring all the other items accounted for by Rosecrans and Felbeck. It was therefore a sort of compromise between the air-standard and the more rational ideal efficiency of the last named authorities. Problem III, as worked out in the paper discussed, started with the assumption of the same initial temperature [mentioned above under (4)] as obtained by Rosecrans and Felbeck. In order to use the temperature-entropy diagram in the way suggested, it would be necessary to test an actual engine, to obtain this initial temperature, by the method of Rosecrans and Felbeck: namely, to analyze the fuel and air, measure the total quantity of each, and obtain the pressure of the gas at the point of initial temperature. It would thus be seen that both of the ideal efficiencies discussed above had limitations in connection with practical testing. It was, in Professor Heck's opinion, preferable to use either the air standard, because it was perfectly definite, or else go the whole length in making complete calculations and tests for the necessary data. When the latter was done, it was always a matter of opinion whether details of the calculations should be based upon actual conditions or ideal. Differences of opinion always occurred when an ideal efficiency of any cycle was modified to take account of departures in the actual engine from the ideal cycle. He wished to suggest that charts be prepared showing constant-pressure and constant-volume curves as presented by the author, for one mol of a "Standard exhaust gas" mixture, the entropy per mol being figured for a stated mixture including CO_2 and H_2O, as well as diatomic gases. Assuming that compression was accomplished upon diatomic gases at room pressure and temperature and the rest of the cycle to be completed upon the standard exhaust-gas chart, might result in a better compromise in the calculation of ideal efficiency. The temperature-entropy diagram presented by the author was the clearest and least confusing chart of the kind that he had seen. An objection to a chart with so few vertical lines was that in using it a fair amount of graphical work—pencil lines or divider measurements—must be done upon the chart itself, soon spoiling the print. This could be met, of course, by having a supply of prints and frequently changing to a new one.

To cover all possible cases the curves should be run to a lower temperature, at least down to 200 deg. fahr.

Two formulas for the specific heat of the diatomic gases compared as follows:

$C_p = 6.93 + 0.0_6 12T^2$	$C_p = 6.933 + 0.0_7 957T^2$
(Goodenough & Felbeck, p. 1329)	(Partington & Shilling, par. 25)

The coefficients of T^2 were essentially as 100 to 80. At 5000 deg. fahr. these gave the values

$$C_p = 10.51 \text{ (G. \& F.) or } 9.74 \text{ (P. \& S.), as 100 to 93}$$

It seemed decidedly preferable, after certain preliminary calculations upon the gas mixtures, to go to the pound of working medium rather than the mol. With oil fuel the weight composition was primary, and the mol of products was a derived quantity. The logical thing was to use the idea of the mol where it offered an advantage, but elsewhere to keep to engineering units. The mean molecular weight of mixture and of products was a very enlightening quantity. Applying his chart to the example quoted by Professor Everett, from Bulletin 150 of the Engineering Experiment Station, University of Illinois, he had found the following:

By the method of his Fig. 12, or by simple use of the chart without correction or adjustment, the efficiency came to 38.1 per cent, which was 2.7 per cent above the computed value of 37.1. The mean molecular weights were, $m_m = 26.62$, $m_p = 27.99$, with the ratio 1.00 to 0.951. The corresponding increase on adiabatic-compression work diminishes the efficiency to 37.4, a discrepancy of less than 1 per cent. Details were, $u_1 = 52$, $u_2 = 162$, $q = 870$, $u_3 = 1032$, $u_4 = 591$, $u_{12} = 110$, $u_{34} = 441$, $w = 331$, $e = 331 \div 870 = 0.381$. The correction changed u_{12} to $110 \div 0.951 = 115.5$ and w to 325.5, so that the efficiency now was $325.5 \div 870 = 0.374$.

In closing, Professor Everett said that with reference to the comments of Professor Heck it might be well to emphasize that the purpose of the paper had been the presentation of the graph of the properties of the diatomic gases whose specific heats could be represented by the equations proposed by Goodenough. With temperature and entropy as coördinates, and using these equations the chart of Fig. 2 had been made. Its usefulness for the graphical solution of problems involving adiabatic or isothermal changes of these gases was obvious.

The question as to whether one was justified in accepting these gases as representative or suitable for the ideal cycle of modern internal-combustion engines was quite another matter, and there was much to be said on both sides. Professor Heck had called attention to some of the drawbacks to such an acceptance, but the author felt that the simplicity gained by assuming the working medium throughout to be air, with its specific heat conforming to the Goodenough equation, warranted its use as a standard in preference to an arbitrarily assumed standard exhaust gas. It was interesting to note that the thermal efficiency for the example quoted in the paper as figured by Professor Heck in accordance with his method and applying all refinements was 37.4 per cent as against the value of 37.1 per cent given in the original paper, a difference quite negligible so far as commercial standards were concerned. Moreover, cycle efficiencies figured on this assumption had all the merits of the Air Standard, but with its chief source of error (assumption of constancy of specific heats) eliminated.

THE MODERN OIL ENGINE

Albert S. Walker[15] pointed out that the double-acting four-cycle engine built by the McIntosh & Seymour Corporation was designed entirely by its own very competent engineers, with little, if any, reference to the experimental work of the Atlas Diesel A-B of Sweden. The V-type railway engines of 200 and 300 b.hp. were an Americanization of, and a decided improvement on, the engines of similar type built by the Atlas Diesel A-B. Fig. 5 in the paper was a cross-section of this small size, 8-in. bore by 9½-in. stroke engine and not of the 1000-b.hp. size. The 1000-hp. size, referred to in line 6 of Table 1, was, however, quite similar in a general way to the engine shown in Fig. 5. The greatest difference between the two engines was that the air and exhaust openings in the cylinder heads of the larger size were on the sides of the engine, slightly increasing the overall width, but decreasing the overall height. The height limitation was more stringent than the width limitation with engines of this size in locomotives. The data in line 6 of Table 1 were entirely correct.

L. H. Morrison[16] thought that it was very unfortunate that more mention had not been made of the Price design, since the two companies in the United States operating under these patents produced more horsepower in engines of this type than did any two other builders of four-stroke-cycle Diesels. Incidentally this patent was the only American Diesel design ever adopted by a European engine builder. The author showed a Falk combustion chamber in Fig. 12 and cited this as a Price design. The Falk combustion chamber was built under the Bannerman patent. The sprays were offset and were not assumed to act as in the Price design.

[15] Superintendent, Experimental Dept., Fairbanks, Morse & Co., Beloit, Wis. Mem. A.S.M.E.

[16] Assoc. Editor, McGraw-Hill Publishing Co., Inc., New York. Assoc. Mem. A.S.M.E.

Edward J. Kunze[17] criticized the author's statement that oil engines using the two-stroke cycle burnt roughly twice the amount of fuel for the same cylinder volume and developed about twice the power. Due to the difficulty of fully charging the cylinder of a two-stroke-cycle engine, the short time of charging, and the poorer scavenging permitted in the usual engine of this type, its volumetric efficiency was much lower than that of a four-stroke-cycle engine. In oil engines this reduction was not so marked as in gas engines because the charge, due to its smaller bulk and higher pressure, was more quickly introduced into the cylinder. The inference made by the author that the Bethlehem Steel Company, through Arthur West, had been the first to solve the scavenging problem by admitting the charge in the center of the cylinder head at which an inlet valve was located, was not correct. In 1904, Louis Illmer, Jr., and Mr. Kunze had shown Mr. West drawings of a 3000-hp. twin-cylinder, two-cycle, double-acting gas engine embodying this feature, which had been included in patent applications filed in 1902. The law that "concurrent flowing gases do not mix except at the fringe of their strata" had been enunciated and verified by Mr. Kunze some years before. Turbulence was not induced merely by swirling the gases unless the introductory velocity of one element of the charge was much greater than that of the other, or if the introduction of one of the gases was by means of high-velocity impinging streams, or again, if obstructions were introduced to produce numerous eddies. Gases flowed smoothly only in channels having unobstructed, uniform cross-sections when they were propelled at constant velocity. If any one of these conditions was neglected, a certain amount of mixing would result. A lip on the end of a piston would direct the flow of the incoming gas or air until the edge of the lip was reached, when the discharge into the combustion space would take the nature of the discharge from an irregular orifice. Some of the fluid, by its inertia, would follow the path in which it was directed, but the stream would tend to spread, especially at the edge of the orifice. Swirling, without doubt, took place in cylinders designed as shown in Figs. 20*a* and 20*b*, but this was accompanied by a mixing of the incoming charge with the products of combustion remaining in the cylinder. Retention of burned products not only displaced some of the incoming charge but, due to the high temperature of these products, also heated this charge, thus expanding it, still further reducing the amount of charge that could otherwise be admitted into the cylinder. Moreover the heating of the incoming charge, when composed of combustible mixture, reduced the compression to which it could be subjected because of danger of premature explosion.

Designs of internal-combustion engines without the above-indicated objectionable features were possible. To accomplish this it would seem better to consider these two functions, mixture and smooth admission of charge, separately instead of trying to solve two antagonistic problems in a single solution. The employment of the double-acting principle would not by any means reduce the weight of a gas engine by one-half. The "double action" type induced a reversal of stresses which had to be provided against in the design by the use of a factor of safety based on the ultimate strength of 20 instead of one of 12 where the stress acted in one direction only.

There were four principal considerations in the design of an engine frame: strength, rigidity, inertia, and casting and metal-forming restrictions. The latter two were not affected by the use of a larger design factor of safety. Certain parts that were made very heavy would not be increased in weight because of the substitution of the double-acting for the single-acting principle. The inertia principle, better understood in the design of machine tools, was important in gas-engine design in that it permitted a more judicious use of metal. In general, it was not necessary to design each element of a frame for strength and rigidity, as casting and metal-forming restrictions would frequently produce sections that had adequate strength and rigidity; but all critical sections of the frame should be checked for strength and rigidity after the layout had been made, and then metal might be added as required to reduce the possibility of undue vibration of the frame.

The Foos Gas Engine Company[18] wrote that their development work had produced a full-Diesel, four-cycle engine with a bore of $8\frac{1}{2}$ in. and a stroke of 11 in. with a speed range of 400 to 900 r.p.m., and covering a power range from 50 to 475 b.hp. The engine was rated on a normal brake m.e.p. of 70 lb. per sq. in., at which pressure the rating at 600 r.p.m. was 33 hp. per cylinder. With the speed increased to 900 r.p.m., and with the same brake m.e.p. the rating was 50 b.hp. per cylinder. The weight of the new Foos Diesel was very much less than the units heretofore accepted as standard design. With ordinary gray-iron-casting construction the weight was from 30 to 60 lb. per hp., depending upon engine size and speed. With the use of steel castings and alloy metal in places where their use was feasible, a further reduction in weight per hp. was secured. The present type was built in sizes of 2 to 8 cylinders. All engines had an overall width of $33\frac{1}{2}$ in. and an overall height of 4 ft. $10\frac{1}{2}$ in. from the mounting flange. The overall length was proportionately small. Outside of the strictly stationary and marine fields, the Foos Diesel had wide application to the portable and semi-portable industrial-equipment field, and in the railway field. The engine would perform heavy-duty service with an economy and reliability of performance not excelled by the larger, slow-speed types. Although the engine operated at speeds three to four times higher than the common type, tests had shown that the fuel economy was equal to that of the average Diesel unit of all classes. Starting out at 400 r.p.m. the fuel consumption at full load was approximately 0.42 lb. per b.hp-hr. increased very slightly with increasing speed. Overload tests had indicated that brake m.e.p.'s as high as 100 lb. per sq. in. were not out of the question. The engine was completely enclosed, there being no moving parts visible. A very simple and effective solid-injection fuel system was employed. The cylinder-head design incorporated the use of dual air-inlet and exhaust valves, with the fuel-injection valve directly in the center of the head. The inlet and exhaust valves were operated with a simple rocker-arm arrangement actuated by a camshaft at the level of the cylinder heads. The camshaft, the fuel-injection pump, and the governor were driven by a silent chain drive from the crankshaft. The crankshaft, the wristpin, and crankpin bearings, as well as all other bearings in the engine, were amply proportioned.

Herman Hugle[19] stated that the Fairbanks-Morse Company had put in high-speed, 2-cycle engines in small sizes, with the 4-cylinder type running at 800 r.p.m. and delivering 60 hp. This engine would be built in 2-, 3-, and 4-cylinder sizes, the bores $5\frac{3}{4}$ in. and the stroke $6\frac{1}{2}$ in. The engine was designed with reverse gear.

The company had not published any figures on the fuel economy but it was as low as 0.48 lb. per hp-hr. at full load at 650 r.p.m.

Warren Viessman[20] pointed out that foreign manufacturers were producing the compressorless Diesel engine, probably to decrease weight and cost. He wished to know whether this had had any material effect on the economy of the engines.

André C. Attendu[21] stated that at the Philadelphia Navy Yard he had for two years been running an experimental engine for the Bureau of Aeronautics, which ran constantly and gave 75 hp. at 1620 r.p.m., with a brake m.e.p. of 72.8. Naturally the 2-cycle type engine had mechanical troubles due to acceleration. He believed that the real high-speed, light-weight engine could be built using very heavy oil, in spite of mechanical difficulties. It was hard to find materials which would stand such high speed and high temperatures. In the engines he had run in Montreal, in England, and in Philadelphia, he had varied speeds from 1500 up to 2300 r.p.m. and the engines had picked up under load wonderfully well. In the first high-speed engine he had had contact with, about six years earlier, he had found that if he left the engine overnight or for a day, one of the first things to do was to clean the valves on account of oxidation. Since he had gone to the solid fuel injection five years ago, at a relatively high pressure, he had found very little trouble of this sort. From experimentation with all sorts of nozzles and different pressures on fuel pumps, he had built a pump with pressures which varied from 500 to 20,000 lb. per sq. in. He had come to the conclusion that the best work was with a larger-size orifice. A $5\frac{1}{2}$-in. bore and 6-in. stroke engine had run

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[18] Springfield, Ohio.

[19] Mechanical Engineer, Fairbanks, Morse & Co., Beloit, Wis. Assoc. Mem. A.S.M.E.

[20] Industrial Power Engineer, Consolidated Gas, Electric Light & Power Co., Baltimore, Md. Jun. A.S.M.E.

[21] Consulting Engineer, Washington, D. C.

hour after hour for two years without any trouble in the injection line. The engine had never been taken down. The reason was that the high velocity prevented carbonization. A two-stage pump was employed. The first stage brought the pressure up to 800 lb. and the final injection was at 6000 lb. per sq. in. By the combination of the high pressure with quite a large nozzle, any kind of fuel could be used without the slightest trouble. The velocity of the piston was actually 2275 ft. per min., the engine running at 2100 r.p.m. He had never had piston or connecting-rod failure due to valve acceleration, adjustment, and braking. In actual commercial work the Buffalo Engine Company had built a large number of engines of 150 hp. for the United States Government, which would run at a maximum of 275 hp. at 1450 r.p.m. Those engines could be driven without a clutch at a weight of 17 lb. per hp.; with the clutch included, at 22½ lb.

The author, Mr. Magdeburger, in his closure answered Mr. Viessman's question regarding fuel consumption. Fuel consumptions with and without the compressor were probably on a par, with possibly a few exceptions. The efficiency of combustion without the air compressor had decreased in most cases, although the development of later engines in Germany had shown that the omission of the air compressor improved the fuel consumption. As far as the experience of the Navy Department was concerned, fuel-consumption figures were in favor of the 4-cycle, as was to be expected, because engines of the 2-cycle type had to have a scavenging pump, which required probably from 8 to 10 per cent of the engine horsepower to drive it.

Regarding Mr. Kunze's discussion, Mr. Magdeburger wrote that when speaking of the two-cycle engines he had referred to the best of them, which showed as high or even higher ratings than standard four-cycle engines, as evidenced by Tables 1 and 2 of the paper, and thus refuted the assertion that they had a "much lower volumetric efficiency."

Heat-Resisting Steels

DURING recent years the interest in heat-resisting steels has grown rapidly and several notable advances in this particular field have been accomplished. However, as Monypenny has recently stated, little has been published with regard to some of the complex steels now being used, which would enable one to compare them with the simpler steels, and thus to judge the intrinsic effect of the extra alloying elements. A review of the work to date would indicate that much fundamental knowledge remains to be determined, and the author trusts that this account of investigations carried out in the Brown-Firth Research Laboratories may be considered a useful contribution to the subject. The researches to be described had a double object. In the first place, they were naturally designed with a view to assisting in the production of better heat-resisting steels; but, in the second place, it was hoped that fundamental data, useful in elucidating the nature of the attack from which steels suffer when exposed to the atmosphere and industrial gases at high temperatures, would be obtained. The achievement of improvement in the characteristics of the steels might, perhaps, be considered a comparatively simple matter of standardizing the approved gaseous influences under standard and controlled conditions of heating, and then submitting to such conditions suitable prepared samples of steels of various compositions and in various conditions. Such, indeed, was the line of attack.

As regards the problem of throwing light upon the nature of the attack at high temperatures, it must at once be conceded that this is extremely difficult. It can be approached from the standpoint of corrosion at ordinary temperatures; and, indeed, in the literature of the subject, it is common to find the rustless and heat-resisting steels dealt with together. The explanation of the phenomenon of resistance, or, to be more accurate, relative resistance at high temperatures, may be similar to that of the phenomenon of rust resistance, the resistance being determined by the formation, or otherwise, and the characteristics of a protective film of a composition determined, in the first place, by the analysis of the steel, and, in the second place, by the composition of the gaseous media. Experimentally, the subject is difficult, if only on one account: great care may be taken in purifying the gases used to produce the atmosphere required, but it should always be remembered that all the steels contain carbon, sulphur, and hydrogen in varying, though in the latter cases small, percentages, and at the high temperatures considered these elements react with the gaseous medium. Thus small quantities of the products which such elements may form will be present, and will require to be considered in any theoretical handling of the results. The importance of such percentages of substances, other than those directly under study, has been well emphasized in the investigations of corrosion at ordinary temperatures by Friend, Evans, and others; it was established by Ariëns Kappers in 1872 that sodium and potassium did not oxidize in dried oxygen, by Cooper in 1882 that chlorine was without action on several metals in the absence of water; while Baker in 1883 established the astonishing fact that purified charcoal could be heated to redness in dried oxygen without burning.

A survey of the data presented results in the detection of several anomalies of such consequence that it is evident that one cannot deduce, from the behavior of the constituent metals and elements, what the response of the alloy produced will be under a given set of experimental or service conditions. As a result of an interesting research in this field, it was stated, for instance, by Utida and Sato that the attack on the alloy was greater as the iron content was increased. The author's data certainly demonstrate that such is not the case; it is shown that some alloys rich in iron have very much greater resistance than other containing less of that element.

The protective action of chromium is generally confirmed. The alloying of chromium with iron in the production of rustless and heat-resisting steels has been well dealt with in technical literature. The patent history of this and other countries also yields much information. The work of Brearley, Haynes, Pasel, Becket, the author, and others laid a sound foundation to the knowledge of that particular field. An interesting paper by McQuigg discusses the influence of chromium in increasing the heat resistance of steels, while the American Society for Testing Materials Symposium of 1924 did much to focus attention on the subject, notable papers being given by Johnson and Christiansen, Hunter and Jones, and Fahrenwald.

The American Symposium has also done much to extend the use of alloys containing nickel in addition to chromium, thus encouraging the utilization of the additive advantage of that element. The apparently anomalous effect of nickel in assisting in the protection of the steel of which it is a constituent may probably be explained by the influence of that element in causing the formation of a particular kind of protective layer; it is certainly an outstanding fact that, with pure nickel in SO_2 there was a heavy attack at 800 deg. cent., but that the attack diminished materially at 900 deg. cent., and became practically negligible at 1000 deg. cent.—W. H. Hatfield in *Engineering*, May 27, 1927, p. 656.

Odor by Specification

AT A recent meeting of the American Chemical Society, at Richmond, Va., there was described a system developed in our laboratories for the close specification of odors.

Briefly, the system recognizes but four constituents of any odor, namely: Fragrant (sweetness of flowers, etc.), Acid (sourness, as of vinegar, camphor, etc.), Burnt (tarry), and Caprylic (goaty—somewhat like putrid).

By comparison with carefully chosen standards, the relative intensity on a scale of nine of each of the four component odor-types present may be closely approximated, and the entire odor expressed as a 4-digit number. For instance, the odor of vanillin (active principle in vanilla extract) is determined as 6021; meaning that it is as *fragrant* as standard 6 of the fragrant series, as nearly free from *acid* as standard zero of the acid series, and as *burnt* and *caprylic* as standards 2 and 1, respectively, in the burnt and caprylic series. A few other similarly specified odors are acetic acid, 3803; citral (from lemon extract), 6645; and oil of cloves, 7642. No odor has yet been found which cannot be adequately defined in this way.

It will soon be possible to specify numerical values and variation limits for the odors of commercial materials and the result will be greater uniformity and satisfaction with such materials.—Industrial Bulletin of Arthur D. Little, Inc., June, 1927.

Worm-Wheel Contact

Preliminary Report of A.S.M.E. Special Research Committee on Worm Gears[1]

By EARLE BUCKINGHAM,[2] CAMBRIDGE, MASS.

The object of this paper is to show how any worm-wheel contact condition can be determined by analysis and to point out in particular the probable influence of the nature of the contact lines between a worm and a worm wheel upon the lubrication conditions, efficiency, and load-carrying ability. Analyses of three helicoids are made and their equations given. The first of these, a convolute helicoid, had its generatrix tangent to a cylinder of any diameter concentric with the axis of the helicoid. The second, a screw helicoid, had its generatrix passing through the axis of the helicoid, and was one limiting case of the first, with the diameter of the base cylinder reduced to zero. The third, an involute helicoid, had its generatrix tangent to a concentric cylinder of such diameter that the helix angle at this diameter was the same as the angle of the generatrix with a plane perpendicular to the axis. The conjugate action of racks is discussed and equations given. An analysis of worm contact is made. Contact lines of screw helicoids used as worms and those of involute helicoids used as worms are discussed. Contact lines of screw helicoids with large helix angles, and involute helicoids with large helix angles, also are treated.

VERY little study of the problem of worm drives makes apparent the difficulties that exist in analyzing the contact conditions between a worm and a worm wheel. The warped surfaces of the worm are not the simplest to deal with. In addition to this, the surfaces of the teeth of the worm wheel, which must be conjugate to the warped surface of the worm, introduce further complications.

The American Society of Mechanical Engineers has recently organized a Special Research Committee on Worm Gears to study this problem and to have tests made under its supervision to obtain reliable experimental data. The first problem confronting this committee is the selection of a rational program for such study and tests, and the present paper may be considered as a preliminary report of the Committee.

In essence, a worm drive is nothing more than a thrust bearing, complicated by the helicoidal surface of the worm, and having line contact instead of surface contact, thus introducing very high pressures and stresses between the contacting lines or minute surfaces.

Worm drives are very often inefficient. The loss of power is caused by friction. The effects of friction are reduced by lubrication. Therefore it seems logical to attack this problem of worm drives as a problem of lubrication. The first step toward this end is to determine the nature of the contact between a worm and a worm wheel as it affects the lubrication. Most of the sliding action in these drives is caused by the rotation of the worm. This sliding is so much in excess of the sliding caused by the conjugate action on the tooth profiles that it would seem as though this conjugate action could be safely neglected, for the present at least.

As regards lubrication, the ideal contact conditions would be to have the contact lines between the worm and the worm wheel as radial lines on the worm, as shown in Fig. 1. This would result in a condition similar to that on a Kingsbury thrust bearing, where a wedge of lubricant is carried along just ahead of the contact line, and this contact line would extend across the worm tooth or thread from top to bottom, thus insuring practically uniform wear over the entire tooth profiles.

As regards lubrication, the poorest contact conditions would be to have the contact lines between the worm and the worm wheel as arcs of circles concentric with the worm, as shown in Fig. 2. Here the lubricant would have but little chance of being renewed as the contact progressed from the bottom to the top of the worm tooth. In addition, the worm tooth would be subject to local wear, and thus tend to lose its form.

As regards the specific stresses in the material set up by such line contact, the conditions shown in Fig. 1 would be much more

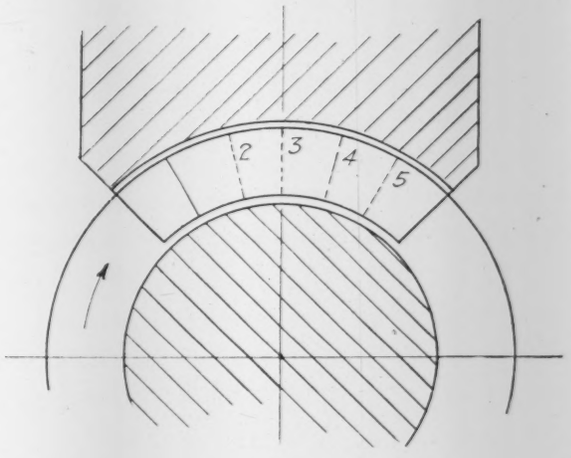

FIG. 1

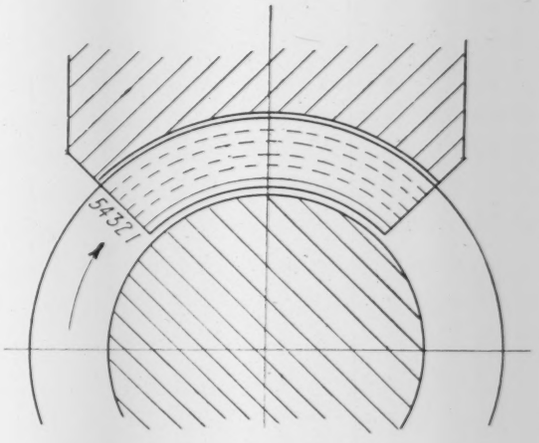

FIG. 2

favorable than those set up by the conditions shown in Fig. 2. In Fig. 1 the line contact would be along a profile with a relatively small radius of curvature, while the point contact in a section perpendicular to the contact line would be between surfaces whose radii of curvature would be relatively large, depending upon the thread angle and the helix angle, often amounting to as much as sixteen or twenty times as large as the average radius of curvature of the tooth profiles. In Fig. 2, on the other hand, the line contact would be along a large radius of curvature, while the point contact would be between small radii of curvature. Thus with equal lengths of contact lines in both cases, a worm drive with the conditions shown in Fig. 1 would have a load-carrying ability about sixteen times as great as a worm drive with the conditions shown in Fig. 2. The length of the contact line under the conditions shown in Fig. 1, however, would only be about one-quarter the length of the contact line under the conditions shown in Fig. 2. Even so, its load-carrying ability would still be about four times as great, together with better conditions of lubrication. We shall

[1] The organization of an A.S.M.E. Special Research Committee on Worm Gears was authorized by the Council in September, 1926, and the A.S.M.E. Main Research Committee appointed Professor Buckingham as its Chairman. While Chairman Buckingham had used the method described in his report with success on two or three critical drives, he presented it not as a recommended practice to be followed in design, but as a subject for investigation and in the hope that its discussion would bring to light a large amount of comparative data which could be used by the Committee in its further study of this subject.

[2] Associate Professor, Standardization and Measurement, Massachusetts Institute of Technology. Mem. A.S.M.E.

Presented at the Annual Meeting, New York, December 6 to 9, 1926, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Slightly abridged.

therefore assume the contact conditions shown in Fig. 1 to be the most desirable ones, and use them as a basis of comparison.

HELICOIDAL SECTIONS

Some years ago the author worked out the equations of the intersection curves and their tangents of certain helicoidal surfaces with planes in various positions, as has been done with a cone in conic sections. All of these helicoids had straight-line generatrices and all of the intersection curves were spirals, corresponding in many respects to the equivalent sections on a cone. The spiral hyperbolas had their asymptotes, etc.

Three helicoids were analyzed. The first had its generatrix tangent to a cylinder of any diameter concentric with the axis of the helicoid. Such a surface would be produced by setting a threading tool with straight cutting edges above or below center, or by tipping the tool off center. This, for want of a better name, will be called a convolute helicoid.

The second had its generatrix passing through the axis of the

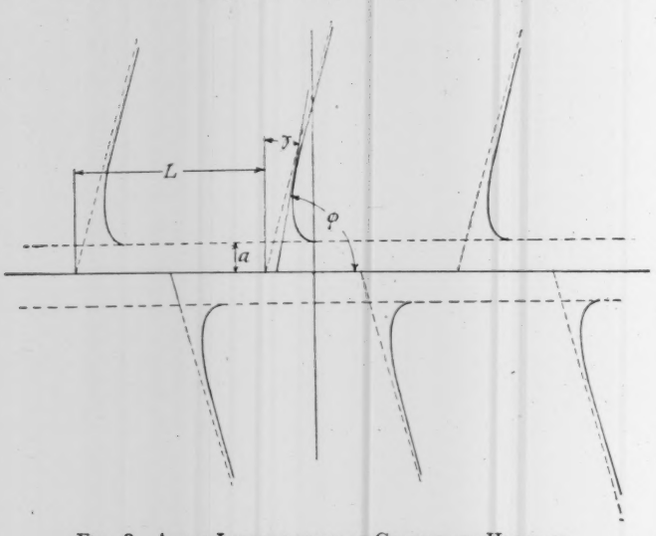

FIG. 3 AXIAL INTERSECTION OF CONVOLUTE HELICOID

helicoid, and was one limiting case of the first, with the diameter of the base cylinder reduced to zero. This is the common screw-thread form and will be called the screw helicoid.

The third had its generatrix tangent to a concentric cylinder of such a diameter that the helix angle at this diameter was the same as the angle of the generatrix with a plane perpendicular to the axis. Or, in other words, the generatrix was a continuation of the helix on the base cylinder. This is another specific form of the first. It is also the form of an involute helical gear and will therefore be called an involute helicoid.

Convolute Helicoid. The following notation will be used throughout, with additional symbols listed as required:

- L = lead of the generatrix, or the distance it advances along the axis of the helicoid in one revolution
- r = any radius of the helicoidal surface; also the length of the radius vector in all polar equations
- γ = angle between the generatrix and a plane perpendicular to the axis of the helicoid
- δ = angle between the tangent to the helix at radius r and a plane perpendicular to the axis—often called the helix angle or lead angle
- θ = vectorial angle
- ϵ = angle of rotation of generatrix
- a = radius of base cylinder to which generatrix is tangent
- D = distance from axis of helicoid to intersecting plane.

The inclination of the generatrix may be in the same direction as the helix on the base cylinder or it may be in the opposite direction. Here we shall consider only the case where this inclination is in the same direction as the helix on the base cylinder.

The equation of the intersection curve of the convolute helicoid with a plane that contains the axis is

$$x = \tan\gamma\sqrt{y^2 - a^2} - \frac{L}{2\pi}\arctan\sqrt{\frac{y^2 - a^2}{a^2}} \quad [1]$$

The equation of the tangent to this intersection curve is

$$\tan\phi = \frac{dy}{dx} = \frac{2\pi y\sqrt{y^2 - a^2}}{2\pi y^2\tan\gamma - aL} \quad [2]$$

The form of this intersection curve is shown in Fig. 3.

The equation of the intersection curve of the convolute helicoid with a plane perpendicular to its axis is as follows:

$$\theta = \frac{2\pi\tan\gamma}{L}\sqrt{r^2 - a^2} - \arctan\sqrt{\frac{r^2 - a^2}{a^2}} \quad [3]$$

The equation of the tangent to this intersection curve is

$$\tan\psi = \frac{rd\theta}{dr} = \frac{2\pi r^2\tan\gamma - aL}{L\sqrt{r^2 - a^2}} \quad [4]$$

The form of this intersection curve is shown in Fig. 4.

The equation of the intersection curve of the convolute helicoid

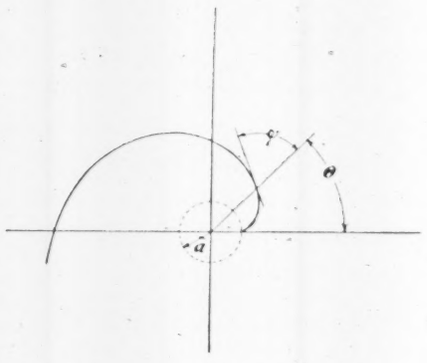

FIG. 4 INTERSECTION OF CONVOLUTE HELICOID WITH PLANE PERPENDICULAR TO AXIS

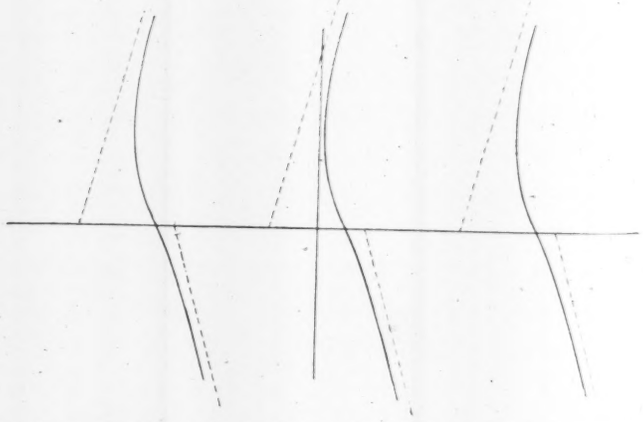

FIG. 5 INTERSECTION OF CONVOLUTE HELICOID WITH PLANE PARALLEL TO AXIS

with a plane parallel to the axis and at a distance D from the axis is

$$x = \tan\gamma\sqrt{D^2 - a^2 + y^2} - \frac{L}{2\pi}\arccos\frac{ay \pm D\sqrt{D^2 - a^2 + y^2}}{D^2 + y^2} \quad [5]$$

The equation of the tangent to this intersection curve is

$$\tan\phi = \frac{dy}{dx} = \frac{2\pi(D^2 + y^2)\sqrt{D^2 - a^2 + y^2}}{2\pi y\tan\gamma(D^2 + y^2) - L[ay \pm D\sqrt{D^2 - a^2 + y^2}]} \quad [6]$$

The form of this intersection curve is shown in Fig. 5.

The limit of conjugate tooth action on these helicoidal surfaces is reached when the tangent to the intersection curve of the helicoid with a plane parallel to the axis is equal to infinity. This would give a rack form with a pressure angle of zero degrees. This limit of conjugate tooth action would be established by the

projection of the points where $dy/dx = \infty$ on a plane perpendicular to the axis of the helicoid. The projection of these points may be determined by the following equations:

$$\sin \epsilon' = \frac{-\pi D \tan \gamma \pm \sqrt{(\pi D \tan \gamma)^2 - 2\pi a L \tan \gamma + L^2}}{L} \quad [7]$$

$$x = D \quad [8]$$

$$y = \frac{D \sin \epsilon' + a}{\cos \epsilon'} = \frac{x \sin \epsilon' + a}{\cos \epsilon'} \quad [9]$$

This curve is plotted in Fig. 6.

Screw Helicoid. The screw helicoid, as defined before, has a generatrix which passes through the axis of the helicoid. Thus it is one limiting example of the convolute helicoid with the size of the base cylinder reduced to zero.

The intersection curve of the screw helicoid with a plane that contains the axis will represent the successive positions of the generatrix 180 degrees apart. Substituting the value of $a = 0$ in Equation [1], we get the equation of this intersection curve as follows:

$$x = y \tan \gamma - \frac{L}{2\pi} \text{ arctan } \infty \quad [10]$$

The arc whose tangent is infinity may be $\pi/2$, $3\pi/2$, $5\pi/2$, etc.

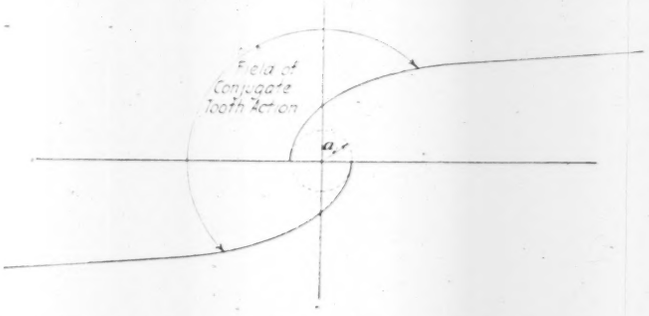

FIG. 6 LIMITS OF CONJUGATE TOOTH ACTION ON CONVOLUTE HELICOID

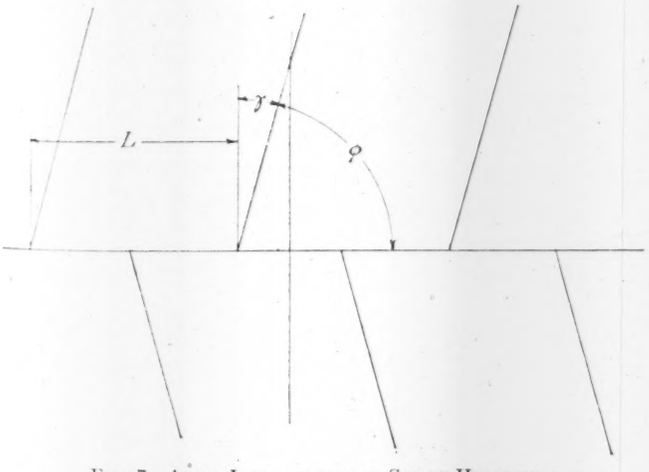

FIG. 7 AXIAL INTERSECTION OF SCREW HELICOID

This curve is composed of a series of straight lines whose tangent is given by the following equation:

$$\tan \phi = \frac{dy}{dx} = \frac{1}{\tan \gamma} \quad [11]$$

This intersection curve is shown in Fig. 7.

The equation of the intersection curve of the screw helicoid with a plane perpendicular to the axis is

$$\theta = \frac{2\pi r \tan \gamma}{L} \quad [12]$$

This curve is an Archimedes spiral. The equation of its tangent is

$$\tan \psi = \frac{r d\theta}{dr} = \frac{2\pi r \tan \gamma}{L} \quad [13]$$

But $2\pi r/L$ = the cotangent of the helix angle at radius r or $1/\tan \delta$, whence

$$\tan \psi = \frac{\tan \gamma}{\tan \delta} \quad [14]$$

The form of this intersection curve is shown in Fig. 8.

The equation of the intersection curve of the screw helicoid with

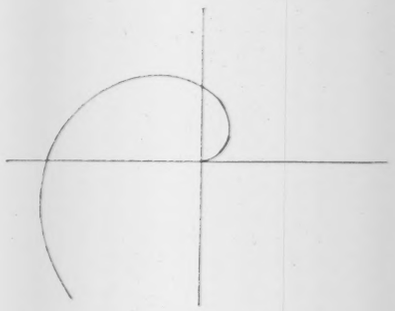

FIG. 8 INTERSECTION OF SCREW HELICOID WITH PLANE PERPENDICULAR TO AXIS

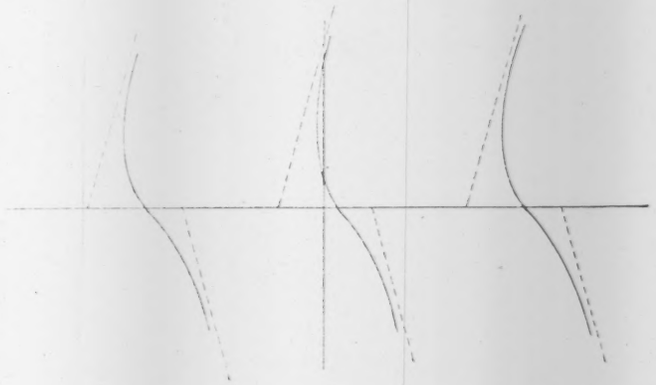

FIG. 9 INTERSECTION OF SCREW HELICOID WITH PLANE PARALLEL TO AXIS

a plane parallel to the axis at a distance D from the axis is

$$x = \tan \gamma \sqrt{D^2 + y^2} - \frac{L}{2\pi} \text{ arctan } \frac{y}{D} \quad [15]$$

The equation of the tangent to this intersection curve is

$$\tan \phi = \frac{dy}{dx} = \frac{2\pi(D^2 + y^2)}{2\pi y \tan \gamma \sqrt{D^2 + y^2} - LD} \quad [16]$$

The form of this intersection curve is shown in Fig. 9.

The projection of the points where $dy/dx = \infty$, or the limit of conjugate action, on a screw helicoid, on a plane perpendicular to the axis may be plotted by means of the following equations:

$$\sin \epsilon' = \frac{-\pi D \tan \gamma \pm \sqrt{(\pi D \tan \gamma)^2 + L^2}}{L} \quad [17]$$

$$x = D \quad [18]$$

$$y = D \tan \epsilon' = x \tan \epsilon' \quad [19]$$

This curve is plotted in Fig. 10.

Involute Helicoid. The involute helicoid, as defined before, is one whose base cylinder is of such diameter that the helix angle there is equal to the angle of inclination of the generatrix. This is another limited example of the convolute helicoid where $\tan \gamma =$

L $2\pi a$. The equations of the intersection curves of the involute helicoid may be determined by substituting this value of tan γ in the various equations of the convolute helicoid.

The equation of the intersection curve of the involute helicoid with a plane that contains the axis is as follows:

$$x = \frac{L}{2\pi}\left[\frac{\sqrt{y^2 - a^2}}{a} - \text{arctan}\,\frac{\sqrt{y^2 - a^2}}{a}\right] \quad [20]$$

The equation of the tangent to this intersection curve is

$$\tan \phi = \frac{dy}{dx} = \frac{2\pi a y}{L\sqrt{y^2 - a^2}} \quad [21]$$

The form of this intersection curve is shown in Fig. 11.

The equation of the intersection curve of the involute helicoid with a plane perpendicular to the axis is

$$\theta = \frac{\sqrt{r^2 - a^2}}{a} - \text{arctan}\,\frac{\sqrt{r^2 - a^2}}{a} \quad [22]$$

This is the polar equation of the involute which is a uniform-rise spiral. The equation of the tangent to this curve is

$$\tan \psi = \frac{r d\theta}{dr} = \frac{\sqrt{r^2 - a^2}}{a} \quad [23]$$

The form of this intersection curve is shown in Fig. 12.

The equation of the intersection curve of the involute helicoid

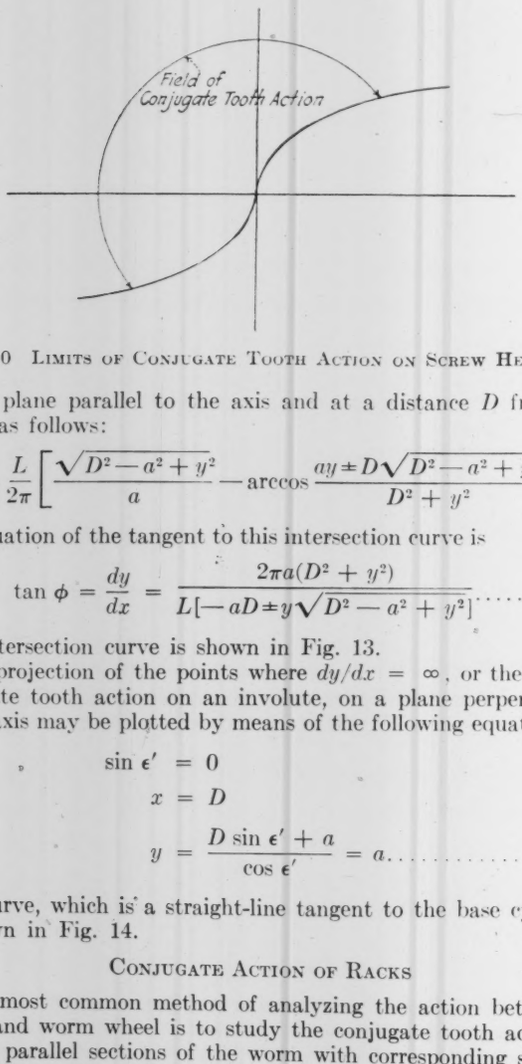

Fig. 10 Limits of Conjugate Tooth Action on Screw Helicoid

with a plane parallel to the axis and at a distance D from the axis is as follows:

$$x = \frac{L}{2\pi}\left[\frac{\sqrt{D^2 - a^2 + y^2}}{a} - \arccos\frac{ay \pm D\sqrt{D^2 - a^2 + y^2}}{D^2 + y^2}\right] \quad [24]$$

The equation of the tangent to this intersection curve is

$$\tan \phi = \frac{dy}{dx} = \frac{2\pi a(D^2 + y^2)}{L[-aD \pm y\sqrt{D^2 - a^2 + y^2}]} \quad [25]$$

This intersection curve is shown in Fig. 13.

The projection of the points where $dy/dx = \infty$, or the limit of conjugate tooth action on an involute, on a plane perpendicular to the axis may be plotted by means of the following equations:

$$\begin{aligned} \sin \epsilon' &= 0 \\ x &= D \\ y &= \frac{D \sin \epsilon' + a}{\cos \epsilon'} = a \end{aligned} \quad [26]$$

This curve, which is a straight-line tangent to the base cylinder, is shown in Fig. 14.

Conjugate Action of Racks

The most common method of analyzing the action between a worm and worm wheel is to study the conjugate tooth action of several parallel sections of the worm with corresponding sections of the worm wheel. The determination of both the resulting rack profiles on the sections of the worm and also their lines of conjugate action is ordinarily accomplished by geometric layouts.

The preceding equations of helicoidal sections enable these rack profiles to be determined to any required degree of accuracy by analysis without the necessity of making enlarged and accurate layouts. The equation for the line of conjugate action is readily determined and may also be used instead of layouts when the equation of the rack profile is known. These equations for the line of conjugate action are as follows:

When x = abscissa of profile
y = ordinate of profile measured from axis of helicoid
R = distance from axis of helicoid to pitch plane of worm (or pitch radius of worm)
ϕ = tangent to profile measured from the axis of x
x' = abscissa of line of conjugate action, and
y' = ordinate of line of action measured from the pitch line

we know that the normal to the tooth profiles at the point of

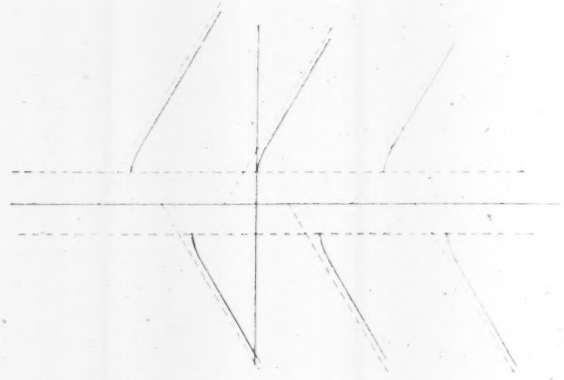

Fig. 11 Axial Intersection of Involute Helicoid

contact must pass through the pitch point. Thus, when the pitch point is the origin of the coördinate system, we have

$$y' = y - R \quad [27]$$

$$x' = -y' \tan \phi \quad [28]$$

Analysis of Worm Contact

As noted in the introduction, in many respects the conjugate tooth action between a worm and worm wheel is of secondary importance. Such action must exist, of course, but its exact nature has but little influence on the conditions of lubrication. It would seem to be of more importance to determine the positions of the actual contact lines between the worm and worm wheel at various positions to see whether these lines approach the conditions shown in Fig. 1 or those in Fig. 2.

There are three ways in which this may be accomplished. First, by determining the profiles of several parallel sections of the worm and their lines of conjugate action by geometric layouts, then using these layouts to determine the projection of the actual contact line on an end section of the worm. Second, by plotting the profiles and their lines of conjugate action in the several parallel sections, using the preceding equations to determine the several coördinates, and then using these graphs or layouts as before to establish the projection of the actual contact line on an end section of the worm. Third, determining directly, by analytical means, the coördinates of the projection of the contact line on the end section of the worm.

Contact Lines of Screw Helicoid Used as a Worm

We shall consider first the contact lines of a screw helicoid used as a worm. In order to simplify the problem, we shall consider this worm as meshing with a worm wheel of infinite diameter, ignoring for the present the questions of interference and undercutting, and other similar limitations to the field of contact. Eventually these questions must be answered in the same manner here as they are for other conjugate surfaces, such as spur gears, etc. The diameter of the worm wheel has no influence on the forms or positions of these contact lines, the larger worm wheel simply using more of them.

The contact lines of a worm with its worm wheel can be determined in the same manner as the contact lines of a rack of vary-

ing profile with a gear. At any moment an infinitely small part of the motion can be considered as a turning motion about the contact line of the pitch cylinder of the worm wheel and the pitch plane of the worm. This contact line, when considered as the axis of this turning motion, will therefore be called the momentary center of the motion.

In order for the worm to be able to turn about this momentary center relatively to the worm wheel, the normal line to any contact point between the worm surface and the worm-wheel surface must pass through this momentary axis. Otherwise it would be impossible for the worm to turn, or rock, slightly about the momentary axis in both directions. This is another way of expressing the fact that, for conjugate tooth action, the normal to the tooth profile at the point of contact must pass through the pitch point.

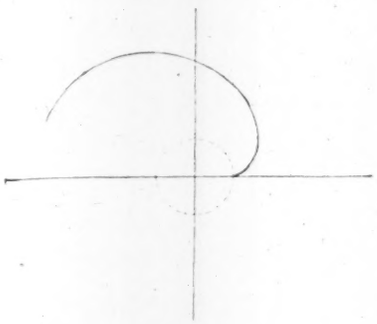

FIG. 12 INTERSECTION OF INVOLUTE HELICOID WITH PLANE PERPENDICULAR TO AXIS

The contact line for any position of the worm can be determined, therefore, by drawing through any point of the momentary axis a line normal to the surface of the worm, and by determining the intersection point of this normal line with the worm surface. Every such intersection point will be a contact point.

The normal to the surface of a helicoid must be perpendicular to its generatrix, also perpendicular to the helix angle at the point of intersection, and also perpendicular to the tangent of any intersection curve at its point of intersection.

Referring to Fig. 15, the position of the screw helicoid relative to the drawing plane can be determined from the location of its intersection profile with the drawing plane. In this case the drawing plane is perpendicular to the axis of the helicoid and the intersection curve is the Archimedes spiral given by Equation [12] as follows:

$$\theta = \frac{2\pi r \tan \gamma}{L}$$

This intersection curve can be located in any position. For convenience, Equation [12] can be plotted directly. Referring to Fig. 15, let

L = lead of generatrix
γ = angle of generatrix
θ = angle of rotation of generatrix
h = length along radial line from momentary axis to intersection curve of screw helicoid
k = distance of projection of contact point along radial line from momentary axis
R = distance of pitch plane from axis of screw helicoid.

Then

$$h = r - \frac{R}{\sin \theta} \quad \ldots\ldots [29]$$

$$k = (h - k)\left[\tan^2 \gamma - \frac{L \tan \gamma \cos \theta}{2\pi(k \sin \theta + R)}\right] \quad \ldots\ldots [30]$$

Solving Equation [30] for k, this develops into a very long equation. For simplification in handling, therefore, we shall let

$A = 2\pi \sin \theta$
$B = 2\pi R - L \sin \gamma \cos \gamma \cos \theta - 2\pi h \sin^2 \gamma \sin \theta$
$C = h(2\pi R \sin^2 \gamma - L \sin \gamma \cos \gamma \cos \theta)$

Then

$$k = \frac{-B + \sqrt{B^2 + 4AC}}{2A} \quad \ldots\ldots [31]$$

When θ = 90 deg., Equation [30] reduces to the form

$$k = h \sin^2 \gamma \quad \ldots\ldots [32]$$

When h is plus, the value of k should also be plus, and the point of contact is above the pitch line; when h is minus, k should also be minus, and the contact will be below the pitch line. If the solution of Equation [31] gives a plus value of k for a minus value of h, it indicates that the contact point in question is outside of the field of conjugate tooth action.

Although the solution of Equation [31] is quite involved, it will be noted that in large measure the intermediate values contain many constants, so that the actual labor of solving for a series of values will be found to be very much reduced after the first set is solved.

Study of Fig. 15 will make it apparent that the contact line will always lie between the momentary axis and the intersection curve of the helicoid, its position depending upon the lead of the helicoid, the angle of the generatrix, and the position of the pitch plane. The smaller this lead and angle, the closer the contact line will be to the momentary axis, or intersection of the pitch plane with the drawing plane.

To obtain the contour of any contact line for any given position of the helicoid in relation to the momentary axis, the preceding equations are solved for several positions of the radial planes, or different values of θ.

In order to determine the change in position of the contact line as the worm is revolved, these equations are solved for successive

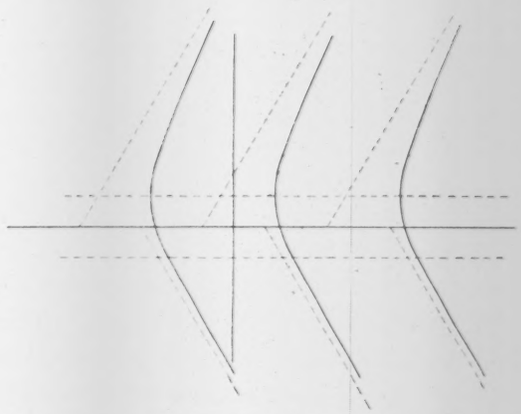

FIG. 13 INTERSECTION OF INVOLUTE HELICOID WITH PLANE PARALLEL TO AXIS

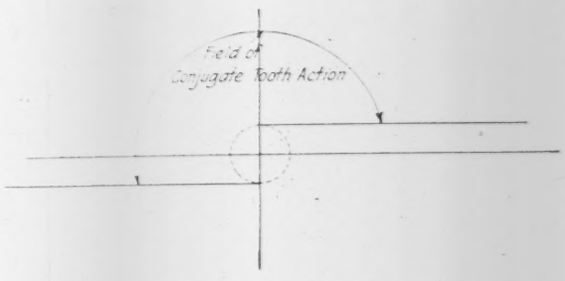

FIG. 14 LIMITS OF CONJUGATE TOOTH ACTION ON INVOLUTE HELICOID

positions of the momentary axis in relation to the helicoid. This is accomplished by determining a new series of values for h, corresponding to the changed positions of the helicoid. For example, if the helicoid is revolved one-quarter of a revolution, all of the original values of h will be altered equally an amount equal to $(L \div 4) \cot \gamma$. The several successive contact lines can then be plotted, which will show the nature of the action as it affects the conditions of lubrication. The actual duration of contact can be determined by establishing the turning angle of the helicoid which carries the contact lines across the face of the worm wheel. This, of course, will also depend upon the diameter of the worm wheel, the duration of contact being greater with worm wheels of larger diameter.

If projections of these contact lines are desired in any other plane, they can be obtained readily by projecting the contact points from the given plane and locating them at a distance equal to $(h - k) \tan \gamma$ from the projection of the momentary axis.

CONTACT LINES OF INVOLUTE HELICOID USED AS A WORM

The contact lines of an involute helicoid used as a worm are much easier to determine. The nature of an involute helicoid is such that every perpendicular to its surface is tangent to the base cylinder. Thus, if we consider a series of intersecting planes tangent to the base cylinder, these planes will contain both the generatrix and the normal to the helicoidal surface. This permits a very simple solution of the position of the contact points. Referring to Fig. 16, let

L = lead of generatrix
γ = angle of generatrix
ϵ = angle of rotation of generatrix
a = radius of base cylinder

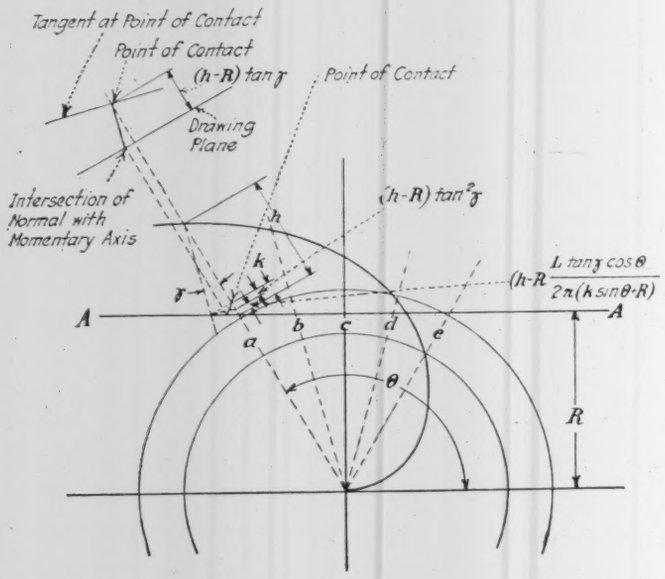

FIG. 15 CONTACT ON SCREW HELICOID

R = distance of pitch plane from axis of helicoid
h = length along line tangent to base cylinder from momentary axis to intersection profile
k = distance of projection of contact point along line tangent to base cylinder from momentary axis.

Then

$$h = a\epsilon - \frac{R - a\cos\epsilon}{\sin\epsilon} \quad \ldots\ldots [33]$$

$$k = h\sin^2\gamma \quad \ldots\ldots [34]$$

Equation [34] holds true for all positions of the planes tangent to the base cylinder.

In order to determine the change in position of the contact line as the worm is revolved, the new length of h is multiplied by the constant $\sin^2\gamma$. This new length of h increases uniformly on all tangent planes. For a turning movement of one revolution it increases an amount equal to the circumference of the base cylinder, or $2\pi a$, and for all other movements in proportion.

If the projection of the contact line is desired in any other plane, it can be projected in the same manner as in the case of the screw helicoid, and the contact points would be located at a distance of $(h - k)\tan\gamma$ from the projection of the momentary axis as before.

We are now in a position to determine the position of the contact lines between a worm and a worm wheel to see whether they approach the conditions of film lubrication, as shown in Fig. 1, or conditions of semi-lubricated surfaces, as shown in Fig. 2. Space here is too limited to do more than examine a very small part of the possible field. We shall therefore confine our attention to a worm of large helix angle, about 45 deg. We shall use as our first example a worm of the form of a screw helicoid.

CONTACT LINES ON SCREW HELICOID WITH LARGE HELIX ANGLE

We shall use as this first example a worm of 3 in. pitch diameter, 8 in. lead with 8 threads or starts, pitch 1 in., and with an included angle of thread of 60 deg. This worm will be of the form of a screw helicoid. This gives us the following values:

L = 8.000
γ = 30 deg.
R = 1.500

Transposing Equation [12], we have

$$r = \frac{\theta L}{2\pi\tan\gamma}$$

With this transposed equation and Equations [29] to [32], and values of θ from 45 to 135 deg., we get the following tabulation:

Original Position of Worm

θ =	45 deg.	60 deg.	75 deg.	90 deg.	105 deg.	120 deg.	135 deg.
r =	1.73205	2.30940	2.88675	3.46410	4.04145	4.61880	5.19615
h =	−0.38927	+0.57735	+1.33384	+1.96410	+2.48854	+2.88675	+3.07483
k =	+0.00489	+0.05018	+0.24348	+0.49102	+0.73532	+0.95128	+1.10525

This contact line is plotted as the first contact line in Fig. 17, together with the intersection curve of the screw helicoid in this original position. When θ = 45 deg., the sign for k is different from that for h, which indicates that this contact is outside of the field of conjugate tooth action. This first contact line is just at the edge of the field of contact between the worm and the worm wheel. We shall therefore turn the worm back in the direction

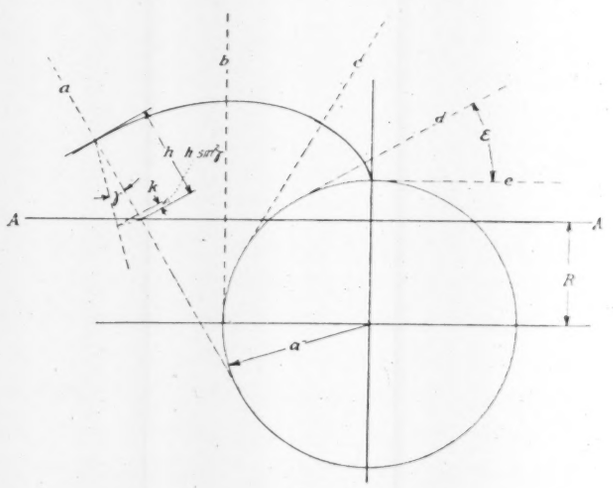

FIG. 16 CONTACT ON INVOLUTE HELICOID

shown by the arrow in Fig. 17 to several successive positions, and establish the forms of other contact lines. This worm has 8 starts, so that a turning movement of 45 deg. would move the thread form the distance of one pitch or tooth interval along the axis of the helicoid. We shall use successive turning movements of 22½ deg., which will correspond to a linear advance of one-half a tooth interval, or ½ in. in this case. Proceeding as before, we obtain the following tabulations:

Second position of worm (turned back 22½ deg.)

θ =	60 deg.	75 deg.	90 deg.	105 deg.	120 deg.	135 deg.
h =	−0.28867	+0.46782	+1.09808	+1.62252	+2.02073	+2.20881
k =	−0.02264	+0.08206	+0.27452	+0.48777	+0.68175	+0.81414

Third position of worm (turned back 45 deg.)

h =	−1.15470	−0.39821	+0.23205	+0.75649	+1.15470	+1.34278
k =	−0.08172	−0.06659	+0.05801	+0.23248	+0.40112	+0.51014

Fourth position of worm (turned back 67½ deg.)

h =	−2.02072	−1.26423	−0.63397	−0.10953	+0.28868	+0.47676
k =	−0.12953	−0.19985	−0.15849	−0.03467	+0.10416	+0.18810

Fifth position of worm (turned back 90 deg.)

h =	−2.88675	−2.13026	−1.50000	−0.97556	−0.57735	−0.38927
k =	−0.16833	−0.31589	−0.37500	−0.32231	−0.21963	−0.16141

Sixth position of worm (turned back 112½ deg.)

h =	−3.75278	−2.99629	−2.36603	−1.84159	−1.44338	−1.25530
k =	−0.20003	−0.41411	−0.59151	−0.65565	−0.59812	−0.55945

All of these contact lines are also plotted in Fig. 17. We shall also determine the limit of conjugate tooth action by means of Equations [17], [18], and [19]. These equations give us the following tabulation:

x =	0.5000	1.0000	1.5000	2.0000
y =	0.99234	1.32718	1.53920	1.68614

These values are also plotted in Fig. 17, and show that when $\theta = 45$ deg. there can be no conjugate tooth action along this plane on the working face of this combination.

A study of Fig. 17 shows that contact lines 5 and 6 are fairly good as regards conditions of lubrication, because they extend from top to bottom of the worm thread at an angle to a concentric circle on the worm. The remaining contact lines tend to follow such concentric circles, however. It is also apparent that the pitch plane seems to have a bad influence on these contact lines, all of them tending to converge toward this pitch plane. Under these circumstances, it should be of interest and value to determine the results of shifting this pitch plane. We shall therefore determine the contact lines on this same worm when the pitch plane is shifted nearer to its axis. Shifting this pitch plane an amount equivalent to adding two more teeth in the worm wheel without changing the center distance, we shall proceed as before. This gives us the following values:

$$L = 8.000$$
$$\gamma = 30 \text{ deg.}$$
$$R = 1.18169$$

Original position of worm (third contact line)

θ =	60 deg.	75 deg.	90 deg.	105 deg.	120 deg.	135 deg.
r =	2.30940	2.88675	3.46410	4.04145	4.61880	5.19615
h =	0.94490	1.66337	2.28241	2.81807	3.25430	3.52499
k =	0.02579	0.27988	0.57060	0.84539	1.09342	1.29556

This contact line is plotted in Fig. 18. In order to cover the face of the worm wheel, it will be necessary to turn the worm back in the direction shown by the arrow in Fig. 18 about four increments as before, and also to turn it ahead in the opposite direction about two of these increments. When turned ahead, it will be necessary to determine the position of the contact points only on the planes where $\theta = 60$ deg., 75 deg., and 90 deg. Thus we get the following values:

First position of worm (turned ahead 45 deg.)

θ =	60 deg.	75 deg.	90 deg.	105 deg.	120 deg.	135 deg.
h =	2.67695	3.39542	4.01446			
k =	0.12232	0.62804	1.00362			

Second position of worm (turned ahead 22½ deg.)

h =	1.81093	2.52940	3.14844
k =	0.06294	0.44847	0.78711

Fourth position of worm (turned back 22½ deg.)

h =	0.07887	0.79734	1.41638	1.95204	2.38827	2.65896
k =	0.00175	0.12582	0.35410	0.59785	0.82447	1.00536

Fifth position of worm (turned back 45 deg.)

h =	−0.78816	−0.06869	+0.55036	+1.08602	+1.52225	+1.79294
k =	−0.01462	−0.01003	+0.13759	+0.34175	+0.54375	+0.71241

Sixth position of worm (turned back 67½ deg.)

h =	−1.65418	−0.93471	−0.31566	+0.21999	+0.65622	+0.92691
k =	−0.02632	−0.12477	−0.07892	+0.07189	+0.24531	+0.37905

Seventh position of worm (turned back 90 deg.)

h =	−2.52021	−1.80074	−1.18169	−0.64604	−0.20981	+0.06089
k =	−0.03503	−0.21765	−0.29542	−0.22390	−0.08377	+0.02642

A comparison of the contact lines in Figs. 17 and 18 shows that the conditions of lubrication in Fig. 18 are very much improved over those shown in Fig. 17. It is hoped that definite tests can be made to prove or disprove the truth of such an analysis as this. According to such a paper analysis, the efficiency of the drive shown in Fig. 18 should be greater than that of the drive shown in Fig. 17, and also its load-carrying ability should be materially increased. Samples for such a test would consist of one-worm and two-worm wheels, both hobbed with the same hob and running at the same center distance; the first worm wheel would be of standard proportions, while the second would have two more teeth.

The foregoing analysis gives an indication of the change in the character of the contact lines when the pitch plane is shifted. This change is much more pronounced with large helix angles than with small ones. We shall now direct our attention to the effect of a change in the form of the helicoid on these contact lines, examining for this purpose the contact lines of an involute helicoid used as a worm. We shall use the same lead and diameter as before, and also select an involute helicoid whose tangent to the axial intersection curve at a diameter of 3 in. is the same as that of the screw helicoid just examined.

CONTACT LINES ON INVOLUTE HELICOID WITH LARGE HELIX ANGLE

Referring to Equation [21], we have

$$\tan \phi = \frac{2\pi a y}{L\sqrt{y^2 - a^2}}$$

Transposing this equation to solve for a, we have

$$a = \frac{yL \tan \phi}{\sqrt{4\pi^2 y^2 + L^2 \tan^2 \phi}}$$

We have the following values:

$$y = R = 1.500$$
$$L = 8.000$$
$$\tan \phi = \tan 60 \text{ deg.} = 1.73205$$
$$a = 1.24029$$

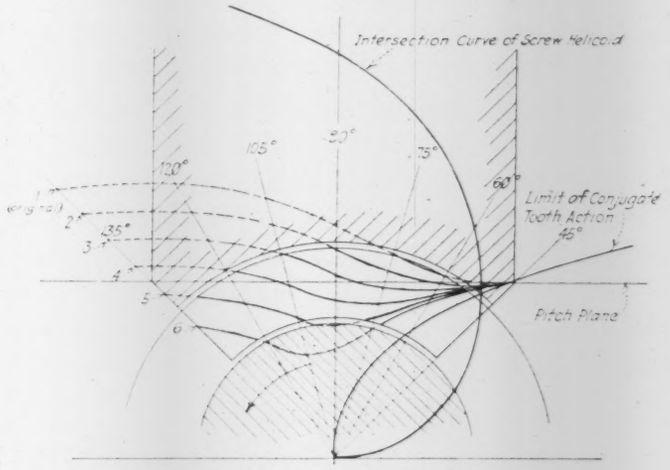

FIG. 17 CONTACT LINES ON SCREW HELICOID OF LARGE HELIX ANGLE

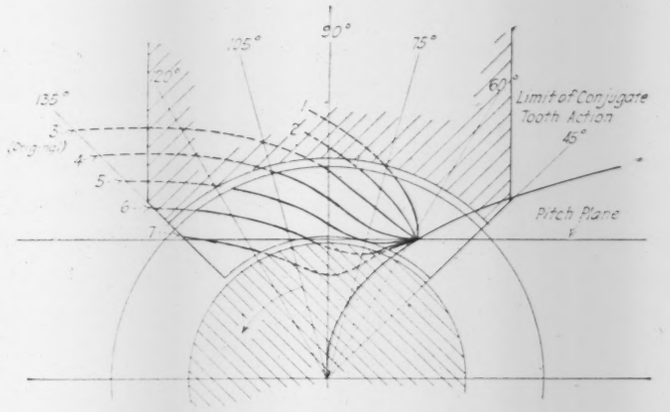

FIG. 18 CONTACT LINES ON SCREW HELICOID OF LARGE HELIX ANGLE

whence

$$\tan \gamma = \frac{L}{2\pi a} = 1.02657$$
$$\gamma = 45 \text{ deg. } 45 \text{ min. } 4 \text{ sec.}$$

Equations [33] and [34] enable the contact points to be determined. Using values of ϵ from 0 to 90 deg., we get the following for the original position of the worm:

ϵ =	15 deg.	30 deg.	45 deg.	60 deg.	75 deg.	90 deg.
h =	−0.84202	−0.20234	+0.09309	+0.28286	+0.40296	+0.44824
k =	−0.43205	−0.10382	+0.04777	+0.14514	+0.20676	+0.23000

These points are plotted in Fig. 19 as position 3, together with the line where $y = a$, which is the limit of conjugate tooth action.

Additional contact points for successive positions of the worm equal to one-half pitch advance, or 22½-deg. turning movements of the worm, are tabulated below:

ϵ =	15 deg.	30 deg.	45 deg.	60 deg.	75 deg.	90 deg.
k_1 =	+0.06777	+0.39600	+0.54759	+0.64496	+0.70658	+0.72982
k_2 =	−0.18214	+0.14609	+0.29768	+0.39505	+0.45667	+0.47991
k_3 =	−0.43205	−0.20382	+0.04777	+0.14514	+0.20676	+0.23000
k_4 =	−0.68196	−0.35373	−0.20214	−0.10477	−0.04315	−0.01991
k_5 =	−0.93187	−0.60364	−0.45205	−0.35468	−0.29306	−0.26982
k_6 =	.....	.....	−0.70196	−0.60459	−0.54297	−0.51973

These values are also plotted in Fig. 19. A comparison of these contact lines with those of Fig. 17 shows very similar characteristics, although their shapes are not exactly alike. This change in the form of the helicoid has had but little influence in this case on the character of these contact lines.

We shall now drop the pitch plane to a distance of 1.18169 in. from the axis of the helicoid as before, and determine the new contact points. This gives us the following values for the original position of the worm:

CONTACT POINTS FOR ORIGINAL POSITION OF WORM

ϵ =	15 deg.	30 deg.	45 deg.	60 deg.	75 deg.	90 deg.
h =	+0.38784	+0.43427	+0.54325	+0.65041	+0.73250	+0.76655
k =	+0.19900	+0.22283	+0.27875	+0.33373	+0.37585	+0.39332

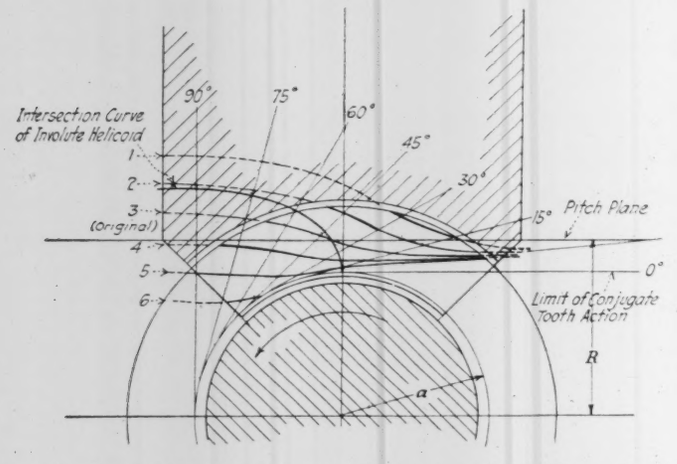

FIG. 19 CONTACT LINES ON INVOLUTE HELICOID OF LARGE HELIX ANGLE

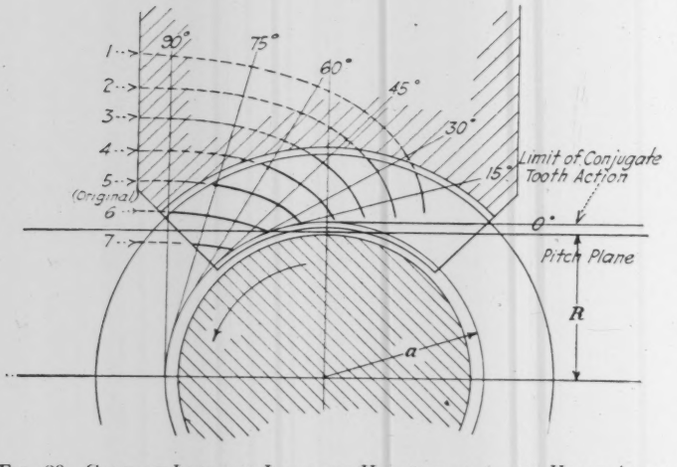

FIG. 20 CONTACT LINES ON INVOLUTE HELICOID OF LARGE HELIX ANGLE

These points are plotted in Fig. 20 as the fifth position of the worm. Additional contact points for successive positions of the worm are determined as before and are tabulated below:

ADDITIONAL CONTACT POINTS FOR SUCCESSIVE POSITIONS OF WORM

ϵ =	15 deg.	30 deg.	45 deg.	60 deg.	75 deg.	90 deg.
k_1 =	+1.19864	+1.22247	+1.27839	+1.33337	+1.37549	+1.39296
k_2 =	+0.94873	+0.97256	+1.02848	+1.08346	+1.12558	+1.14305
k_3 =	+0.69882	+0.72265	+0.77857	+0.83355	+0.87567	+0.89314
k_4 =	+0.44891	+0.47274	+0.52866	+0.58364	+0.62576	+0.64323
k_5 =	+0.19900	+0.22283	+0.27875	+0.33373	+0.37585	+0.39332
k_6 =	.....	−0.02708	+0.02884	+0.08382	+0.12594	+0.14341
k_7 =	.....	.....	−0.22107	−0.16609	−0.12397	−0.10650

These values are also plotted in Fig. 20. The nature of these contact lines is very similar to those in Fig. 18, which indicates that the position of the pitch plane has a greater influence on the characteristics of the contact lines than has the exact form of the helicoid.

It is of interest to note that the FJ worm drive developed by the David Brown and Sons Company of England is an involute helicoidal worm with the pitch plane tangent to the base cylinder and also tangent to the outside diameter of the worm wheel.

Discussion

JOSEPH C. O'BRIEN,[1] commenting on the paper, wrote that in Fig. 1, on a line of contact such as line 3, there would be no conjugate action between the worm thread and wheel tooth. Instead, there would be pure sliding from top to bottom along an inclined plane. There would be a similar, but lesser, condition of sliding on lines 2 and 4, and 5 and 1. That was, the conjugate action tended to increase as the distance of the lines from the worm axial plane increased.

Mr. O'Brien believed that the contact lines in Figs. 18 and 20 were superior to those selected as ideal. Suppose in Fig. 20 that the worm turned in a direction contrary to that shown by the arrow and that contact was at line 7. Transposing line 7 circumferentially $22\frac{1}{2}$ deg. in the direction of the arrow, then the length of worm thread from this point to line 6, measured along the helix, came into action during this rotation. The length of worm tooth from lines 7 to 6, measured along the helix, also came into action, indicating about $33\frac{1}{3}$ per cent rolling action between the helices. This action, as regarded lubrication, would be superior to that selected as ideal. The contact lines of Fig. 20 were the best of the four conditions exhibited, not on account of their similarity to the ideal selected, but rather on account of the uniformity of spacing across the face of the wheel. Such spacing meant uniform progression of contact, proportional to the rotation of the worm. Also, the lines of contact were further apart, which implied for the same increment of rotation a greater percentage of rolling action.

With reference to the best position of the contact axis, it was not wise to neglect entirely the conjugate action of the teeth, particularly when the lead angle approached 45 deg. and a large number of threads were used. Gear ratios as low as 3:1 might be encountered, and lines of contact that were good with respect to the worm axis might be had with respect to the gear axis. The mutual effect of the contact lines should be considered, and also the path of the lines. This might be made clear by reference to Fig. 20. The contact lines at the extreme left approached a position parallel to the wheel axis, which had been characterized as bad in the case of the worm, although the worm turning action was at an angle of 45 deg. Conjugate action was at its maximum at this point. Considering contact line 1, in which the contact approached a plane perpendicular to the wheel axis, there was but little conjugate action, the worm turning action was $22\frac{1}{2}$ deg., and sliding predominated. Whether or not the position of the contact lines could be shifted to improve the general condition was problematic. To obtain the best action in all parts of the contact, it was possible that the ideal lines might vary across the face of the wheel, being affected possibly by the diameter of the worm, the diameter of the wheel, and by other factors.

Referring again to Fig. 20, each contact line represented a movement of the worm through $22\frac{1}{2}$ deg. Therefore, two contact lines in planes of rotation of the wheel, on the working surface of the teeth, represented two teeth in contact. Continuity of tooth action existed only in a small area at the $22\frac{1}{2}$-deg. point to the left of the worm axial plane. The action to the right of the axial plane had barely one-half pitch path of contact. Such a worm apparently would have continuity of tooth action in planes of rotation of the wheel in only about 25 per cent of the width of the wheel face.

From the data in the paper, contact diagrams of single, double, triple, and even quadruple worm threads would appear very poorly in comparison with the diagrams used as illustrations. They apparently would be so similar as to render this method of analysis unsuitable for worms of very low lead angles; that was unsuitable as a method of estimating the capacity of the worm drive.

H. E. MERRITT[2] confirmed the author's surmise that the method of determining the location of the lines of contact was an essential step in worm-gear design. Although in the paper it was suggested that such a detailed geometrical analysis should only be used to obtain information of a general kind on worm-gear design, it was Mr. Merritt's practice to determine the "zone of contact" for every

[1] Pittsburgh Gear and Machine Co., Pittsburgh, Pa.
[2] Research Engineer, David Brown & Sons, Ltd., Huddersfield, England.

worm gear put into production. This applied particularly to the type of worm gear referred to in the paper.

In determining the lines of contact, he wrote, it was essential to determine the location of these lines, not merely on the projected end section of the worm, but also their position both axially with regard to the worm and their circular projection on the wheel tooth surfaces. This was necessary in order to determine the total length of the lines of contact and also the curvature of the "zone of contact" in planes parallel to the worm and normal to the wheel axis. In order to determine the location of the paths of contact he used a geometrical method which did not involve constructing either the worm or wheel tooth profiles, but gave the line of contact directly. This construction took about 20 minutes to complete.

In order to reduce the amount of labor required in designing worm drives, however, the David Brown & Sons' standard system of worm-gear design made use of the principle of geometrical similarity. This system was based upon the use of 40 standard forms of worm thread, each of which had been completely analyzed for "zone of contact." These worms ranged from 1 to 10 threads, and the pitch diameter was a nominal dimension which was made equal to a whole number of modules. This number was denoted by the symbol Q and ranged from 6 to 10.

The zone of contact was, however, only a preliminary stage in determining the load-carrying capacity of worm gearing. Having found the length of the lines of contact, the relative radius of curvature was determined for what was regarded as a mean position, and from this a zone factor was deduced by means of which, given the module, the number of teeth in the wheel, and the tooth load, the surface stress could be computed. It had been found that surface stresses calculated in this way served to form a reasonably accurate basis of design. In cases where failure had occurred, calculation of the surface stresses, based on analysis of the zone of contact, had explained the failure.

L. R. Buckendale[3] wrote that the author's mathematical analysis of the worm-wheel contact conditions substantiated the results of graphical investigations of these same conditions. It also substantiated the hypothetical reason assigned for the evident tendency to pit, which was shown by most applications of the straight involute-type gearing, figured with the conventional addendum and dedendum. The author also confirmed mathematically the radical difference in contact lines obtained by means of the F. J. system. These contact lines, as shown, had been confirmed by placing worms and wheels together and etching to show the lines of contact obtaining at each instant of tooth phase.

George H. Acker[4] wrote that the immediate problem had a slightly different nature from the one suggested by the author, namely, lubrication. In the present state of the art, the real loading limitation was determined by the safe fatigue limit of the bronze, of which it was at present throught that the gear must be composed, rather than any safe lubrication load.

The specific examples of tooth contact in the paper covered only the two extreme cases of tooth form, whereas the actual practice in tooth form might range between these two forms, without actually meeting either. In the type of gear described by the author as "convolute helicoid" gearing, the writer had seen evidences of wear due to lubrication failure, but this wear had, in almost every case, been accompanied by pitting of the bronze. More often, however, the pitting was to be found in some degree, without any indication of undue wear. Mr. Acker therefore concluded that, with this convolute gearing, with the ordinary pitch-plane placement, the ability of the lubricant to carry load exceeded that of the bronze. It should be emphasized that such a conclusion was predicted entirely on use of the correct lubricant.

The author's contact lines, Mr. Acker wrote, were the intersection of the plane of action with the helicoidal worm surface. Could not a broad, thin beam of light be substituted for the contact plane; a highly polished worm for the formula, and a photographic plate, or the eye, for the plot? A desk lamp, two slotted pieces of cardboard, and a worm had served to indicate that the matter was at least worth investigating. If such a method could be developed, it would have one distinct advantage over a mathematical plot, in that, for a fixed thickness of light beam, the relative load-carrying ability of varying surfaces could be roughly gaged by the width of the reflection from the worm.

Thomas W. H. Jeacock[5] in a written discussion stated that the failure of the gear was laid at times to the bronze, at times to the formation of the tooth, and at still other times to the lubricant. In an endeavor to better the whole prescribed 89-11 bronze alloy or "stone metal," as it was called some years ago, he had found that with the increased load the gears were asked to carry, because of the development of motor trucks, the teeth pulled out. In other words, the gear bronze was asked to do things far beyond its physical limitations. In working with different alloys, some results were accomplished with aluminum bronze. It was found that aluminum bronze under certain pressures, where the temperatures ran up, failed. It was then found necessary to go back to the old gear-bronze formula and work from that. Success had been obtained with the introduction of nickel in varying quantities.

In his closure, the author said he believed Mr. O'Brien in his discussion laid too much stress on conjugate tooth action. Conjugate tooth action was not a necessity on worm drives. All of the globoid worms were free from it and many of them had proved satisfactory. Conjugate tooth action was a necessary evil that existed when the standard forms of helicoids were used as worms. This action must be considered when determining the field of contact, and played its part in the determination of effective combinations of helix angles and thread or pressure angles, but its nature was a secondary matter, as most of the sliding between the worm and wheel was caused by the turning movement of the worm.

He was correct in stating that the contact diagrams of single, double, triple, and even quadruple worm threads of conventional designs and proportions showed up very poorly. As a matter of fact, the contact conditions in these cases, as regarded lubrication, were very poor. The author believed that this was one of the most important features of worm and worm-wheel design, and if it should prove true in practice, and by actual experiment, it was possible to modify the design of any of these drives in a simple manner to obtain contact lines of almost any nature that might prove to be most desirable. The A.S.M.E. Special Research Committee on Worm Gears hoped to be able to have tests conducted to determine the influence of the nature of these contact lines on the efficiency and load-carrying ability of worm drives. With such data as a foundation, the next step would be to work up design data so that the most favorable conditions as proved by actual tests would be secured.

Mr. Merritt brought out the fact that it was also necessary to establish the zone of contact as well as the nature of the contact lines. The author's analysis gave the basic equations necessary to establish this zone of contact, but his discussion had been limited by the space available to but a single feature of this involved subject—that of the influence of the contact lines on the lubrication conditions. As Mr. Merritt stated, once a specific type of general design had been adopted, it was possible to work up general data in such form that each individual worm drive might be rapidly analyzed in detail. There were also certain simple approximations which could be used to give an accurate qualitative analysis in a very short and simple manner.

Mr. Acker and Mr. Jeacock called attention to the limitations of the load-carrying ability of the materials and expressed the opinion that this was a more immediate problem to be attacked than that of lubrication. The author could not take any exception to this, but would call attention again to the fact that an improvement in the nature of the contact lines as regarded lubrication also reduced the maximum stresses that were set up in the materials under load, so that a series of experiments along the lines suggested in the paper would be attacking both problems at the same time. Thus if experiments should point the way to a worm and worm-wheel design of higher efficiency and greater load-carrying ability by the same material, such experiments would seem to be worthy of serious consideration. A paper analysis showed that this improvement was possible, and it was hoped that the interested industries would give substantial support to the work of the committee.

[3] Sales Engineer, Timken-Detroit Axle Company, Detroit, Mich.

[4] Chief Engineer, Cleveland Worm and Gear Company, Cleveland, Ohio. Jun. A.S.M.E.

[5] President, Buffalo Bronze Die-Cast Corporation, Buffalo, N. Y.

Distribution of Belt Creep and Slip

By R. F. JONES,[1] ITHACA, N. Y.

This paper presents the results of a study of the distribution of belt creep and slip. The stroboscope slip meter used in the tests and the method followed by the investigators as well as the theory involved are explained. The results show that the arc over which the belt creeps, starts at the last point of contact and extends in the direction opposite to rotation an amount depending on the load. When the arc of creep coincides with the arc of contact, true slip begins. The greatest slip always occurs at the last point of contact.

The amount of load that can be carried by a given belt under given conditions at the merging point of slip and creep varies with the coefficient of friction.

If a small pulley is used with a larger one of the same material the merging point will be reached at the small pulley first, and therefore a majority of the total slip will take place on this pulley. Excessive slip may occur on the small pulley while the belt is creeping below the merging point on the large one. The merging-point load is a function of the pulley diameter, as well as the arc of contact and the coefficient of friction.

SINCE the famous experiments of Wilfred Lewis, there has been a great deal of research on belting problems in this country and abroad. However, there is one phase of this subject which has been neglected, namely, the distribution of slip around the pulley, and there still seems to be an absence of data, especially on domestic belts of standard construction. The purpose of this paper is to describe the author's experiments on slip distribution, using standard domestic belts.

Alexander Fieber[2] used a tachometer with a wheel running on a solid rubber belt to determine the speed at various points throughout its length. The amount of speed variation due to creep along the average belt is usually less than 1½ per cent of the belt speed. A very small error in the tachometer reading, or a normal variation in the speed of the testing machine, can cause a large error in the results. Therefore it was decided that only rough approximations could be obtained by this method.

Description of Apparatus

The scheme finally adopted was designed to measure the speed of the belt at any point relative to the pulley, rather than the actual speed of the belt, by employing the principle of the stroboscope, as illustrated in Fig. 1. Each pulley had a thin galvanized-iron disk A with equally spaced oblong holes punched around its periphery, mounted on its outside edge. The bottom edges of the holes coincided with the edge of the pulley face to allow a clear view of the belt through them.

The portable wheel B, which could be placed against the belt at any point, was of such diameter that its holes approximately coincided with those on the disk A as the pulley turned. It was built of steel with a heavy rim to increase the inertia, and mounted on ball bearings. Most of these experiments were made on 24-in. cast-iron pulleys, or a combination of 6-in. and 24-in. cast-iron pulleys. Therefore if the disk for the 6-in. pulley contained 3 holes, those for the 24-in. pulleys needed 12 holes, while the wheel B, about 6 in. in diameter, also had to have 3 holes. Several wheels B of slightly varying diameter were used, to compensate for small differences in belt thickness and pulley diameter.

White backgrounds C were erected behind each pulley as reflectors, and artificial illumination was provided as needed.

The belt was first brought up to speed (2000 ft. per min. in these tests) at no load, at which creep would be approximately zero, and the wheel was held against the belt at a point about 30 deg. past the point where the belt came on the pulley. Then by looking at the point of apparent intersection of the holes in the disk and wheel from a position two or three feet away from the disk, an oblong of light could easily be seen when the holes came into synchronism. Usually the aperture would be moving slightly one way or the other until it disappeared, indicating that the holes were not quite in synchronism. The rate of travel of the aperture under these conditions was the zero correction.

When the belt was put under load, creep or slip could be measured at any point on the pulley by noting the rate at which the holes synchronized. Every time the holes came into line, the belt at that point had slipped the distance between the holes on the disk measured at the pulley surface, less the zero correction.

The slip may be expressed as an equation by letting

D = pulley diameter in feet
N = number of holes in the disk
X = number of synchronisms in Y seconds
S = slip in feet per minute
S_1 = slip uncorrected for zero error.
S_0 = zero correction.

Then the distance between holes in the disk at the pulley surface is

$$\frac{\pi D}{N} \qquad [1]$$

and the number of synchronisms per minute is

$$\frac{60X}{Y} \qquad [2]$$

S_1 is the product of Equations [1] and [2], and

$$S_1 = \frac{60\pi DX}{NY}$$

As the relation between D and N was always $\frac{1}{6}$ for these tests, the equation was simplified to read

$$S_1 = \frac{10\pi X}{Y}$$

If there are X' synchronisms in Y' seconds at no load,

$$S_0 = \frac{10\pi X'}{Y'}$$

and

$$S = 10\pi\left(\frac{X}{Y} \pm \frac{X'}{Y'}\right)$$

There was some difficulty in obtaining accurate readings at the points near the ends of the arc of contact, because the pitch line was shifting at these positions. Therefore the wheel was placed out on the belt about 12 in., as indicated in positions 1 and 6 of Fig. 1. From this position the line of vision through the holes was angular to the disk instead of perpendicular, but this caused no difficulty. Readings at points 1 and 6 indicated the slip of the belt with respect to the pulley at the very ends of the arc of contact, except for a possible error due to a change in length of the belt during that 12 in. of travel, which was small.

This apparatus may be used to give an approximate value of the total slip or loss of pulley speed in transmission. Assuming that there is no appreciable change in the unit length of the belt as it passes from one pulley to the other, the speed of either strand T_1 or T_2 serves as a measure to determine the peripheral speed of the two pulleys relative to each other. The above assumption is justified by previous experiments in which a test section could be stretched and released as rapidly as would occur on a belt traveling at 5000 ft. per min. under load while the force and elongation were automatically recorded. Total-slip observation obtained by this method agreed closely with simultaneous readings from the regular slip meter, and thus acted as a check on the accuracy of the stroboscopic wheel.

The testing machine is a development of the type originally used by Wilfred Lewis. Two 100-hp. electric cradle dynamometers

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[1] Research Engineer, The Leather Belting Exchange Foundation.

[2] *Zeitschrift des Vereines deutscher Ingenieure*, 1909, p. 1641.

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are used, one to drive the other through the test belt. One is permanently mounted, while the other is placed on ball-bearing wheels which roll on a carefully leveled track. The following measurements may be taken: torque at either shaft, r.p.m. of either shaft, total belt slip, the sum of the tension in the two strands

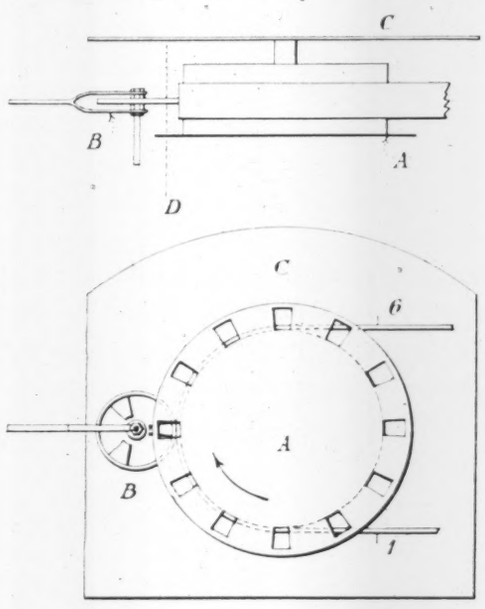

FIG. 1 DIAGRAM OF STROBOSCOPIC SLIP METER

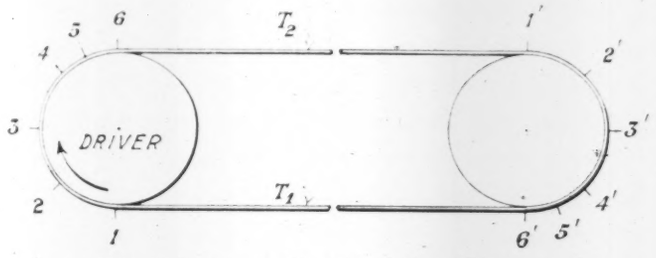

FIG. 2 POINTS AT WHICH SLIP WAS MEASURED

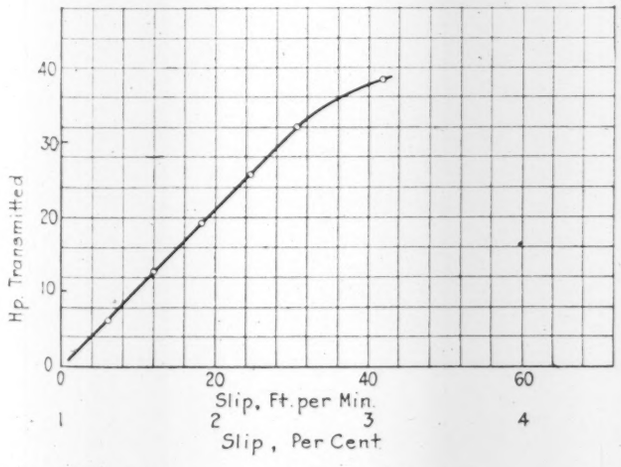

FIG. 3 TOTAL SLIP—BELT NO. 1, 24-IN. PULLEY DRIVING A 24-IN. PULLEY

less centrifugal force, center distance, arc of contact, and belt sag. Center distances up to 25 ft. and shaft speeds up to 1500 r.p.m. can be used.

The belts should be well balanced and of uniform thickness to permit smooth, even running of the wheel B. Therefore two endless 4-in. light double leather belts and one heavy 4-in. single fabric and rubber belt were chosen. The latter was joined with hooks, except for a section about $^3/_4$ in. wide at the center, where the hooks were removed to prevent jumping of the wheel as the lace passed under it. They were approximately 30 ft. long, giving about 12 ft. center distance on the 24-in. cast-iron pulleys.

These three belts varied widely in power-transmitting ability as can be seen from their capacity curves using 24-in. pulleys in Figs. 3, 12, and 15. Their coefficients of friction at 1.5 per cent slip (30 ft. per min.) computed from the formula $\frac{T_1 - T_c}{T_2 - T_c} = e^{f\theta}$ were (No. 1) 0.99, (No. 2) 0.64, and (No. 3) 0.44. Belt No. 1

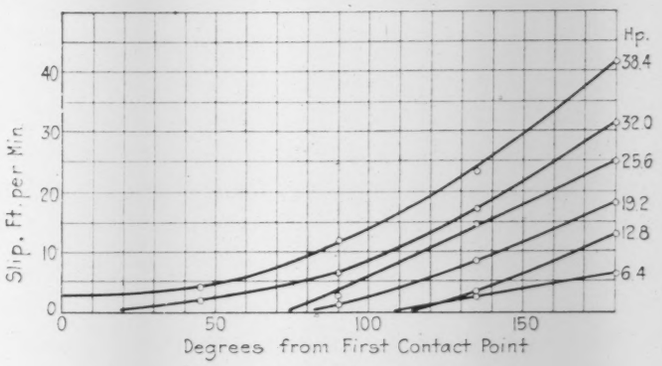

FIG. 4 SLIP DISTRIBUTION—24-IN. DRIVEN PULLEY

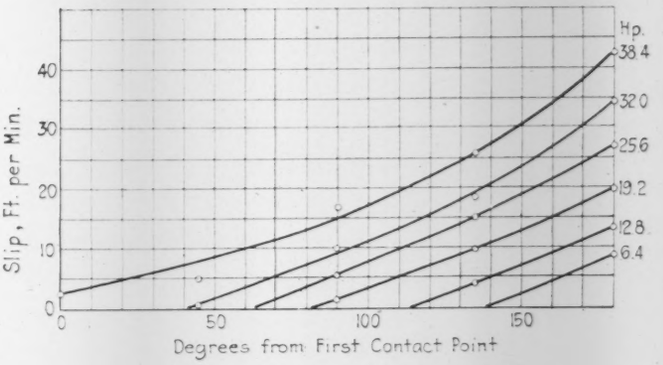

FIG. 5 SLIP DISTRIBUTION—24-IN. DRIVER PULLEY

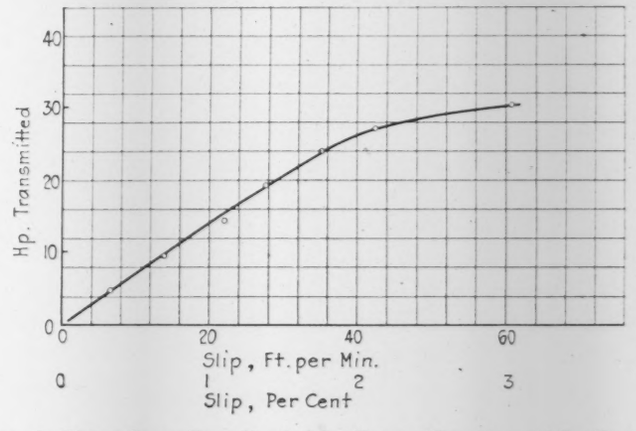

FIG. 6 TOTAL SLIP—BELT NO. 1, 6-IN. PULLEY DRIVING A 24-IN. PULLEY

represents normal leather with a high coefficient of friction, belt No. 2 corresponds closely to a leather belt in need of dressing, and belt No. 3 is a typical example of the fabric and rubber belts.

The load was applied in increments depending on the belt, and the slip measured at the six points on each pulley, indicated in Fig. 2. Points 1, 2, 3, and 4 were approximately 45 deg. apart, while 4, 5, and 6 were from $22^1/_2$ to $26^1/_2$ deg. apart, depending on the arc of contact. The belt speed was 2000 ft. per min. for all belts. The initial tension was set and maintained throughout each experiment at a value giving the smoothest operation, which was 170 lb. per sq. in. for belts Nos. 1 and 2, and 235 lb. for No. 3.

There was one possibility of error, namely, the slip of wheel B on the belt. At positions 2, 3, 4, and 5, where the belt was in

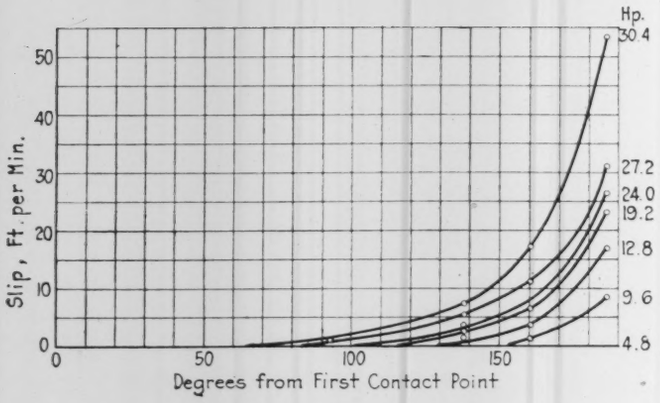

FIG. 7 SLIP DISTRIBUTION—24-IN. DRIVEN PULLEY

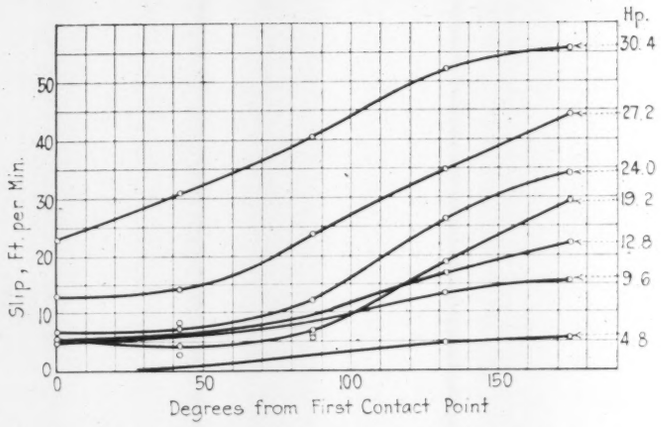

FIG. 8 SLIP DISTRIBUTION—6-IN. DRIVER PULLEY

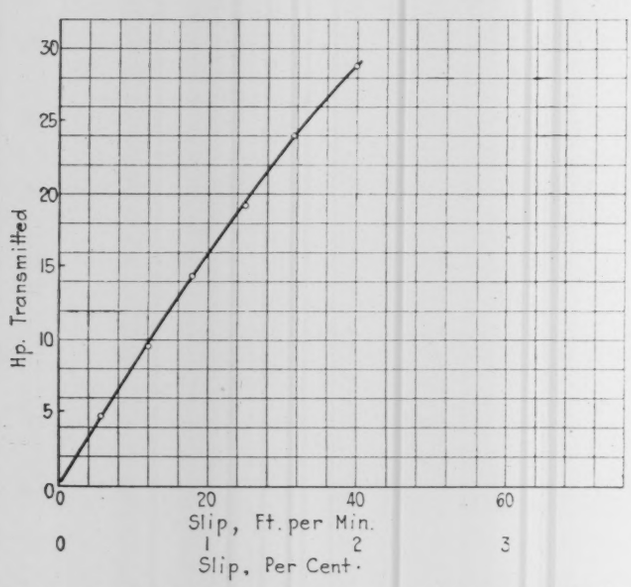

FIG. 9 TOTAL SLIP—BELT NO. 1, 24-IN. PULLEY DRIVING A 6-IN. PULLEY

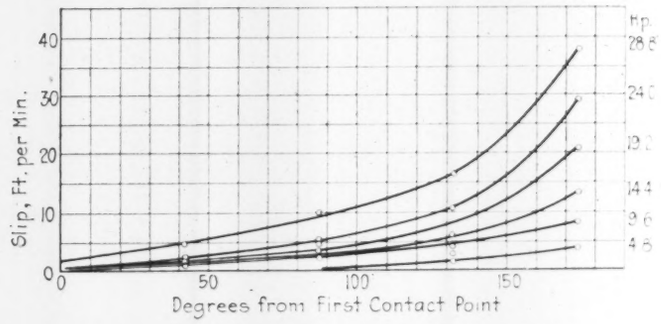

FIG. 10 SLIP DISTRIBUTION—6-IN. DRIVEN PULLEY

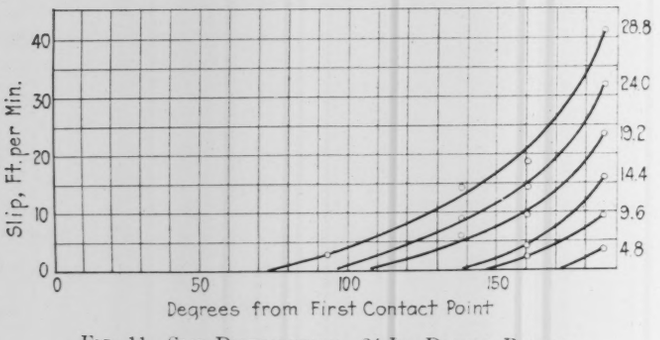

FIG. 11 SLIP DISTRIBUTION—24-IN. DRIVER PULLEY

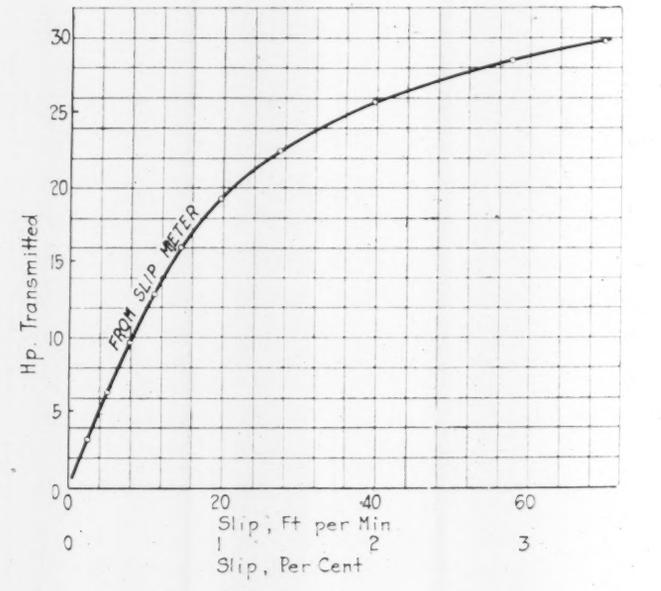

FIG. 12 TOTAL SLIP—BELT NO. 2, 24-IN. PULLEY DRIVING 24-IN. PULLEY

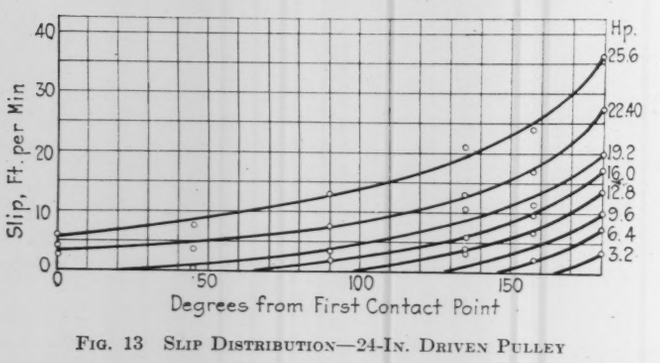

FIG. 13 SLIP DISTRIBUTION—24-IN. DRIVEN PULLEY

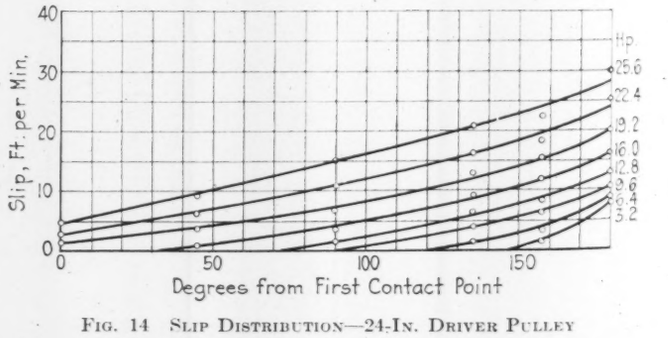

FIG. 14 SLIP DISTRIBUTION—24-IN. DRIVER PULLEY

direct contact with the pulley, several readings at the same point would check each other within 2 per cent. The pressure on the wheel B could also be varied with no appreciable change in the reading. But at positions 1 and 6 the wheel was placed about 12 in. out on the belt, and was therefore subject to more or less vibration, and at times a slight flapping of the belt. Under these conditions the readings usually checked within 5 per cent, because the inertia of the wheel and its ball-bearing mountings kept it running evenly. A check on the accuracy of the readings at points 1 and 6 and at 1′ and 6′ was made by comparing the total slip obtained from

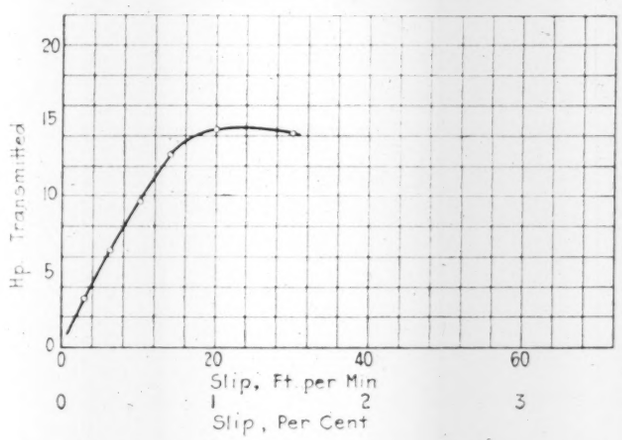

FIG. 15 TOTAL SLIP—BELT NO. 3, 24-IN. PULLEY DRIVING 24-IN. PULLEY

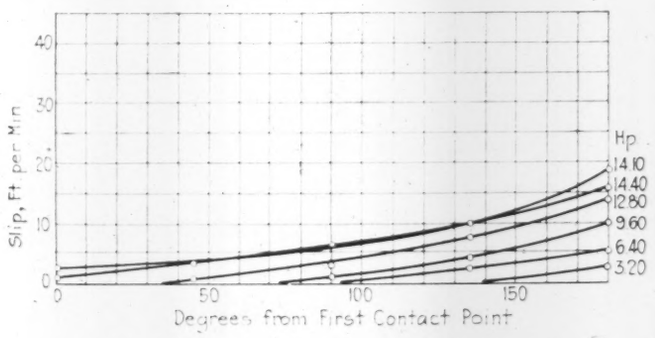

FIG. 17 SLIP DISTRIBUTION—24-IN. DRIVER PULLEY

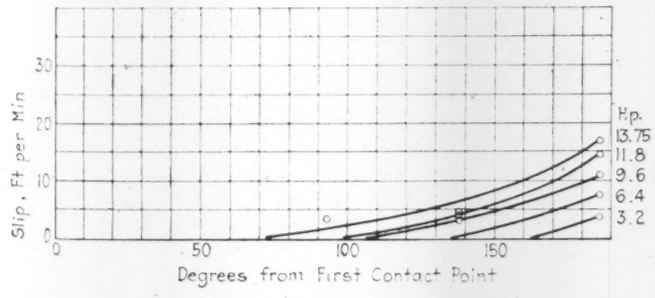

FIG. 19 SLIP DISTRIBUTION—24-IN. DRIVEN PULLEY

them with the slip indicated by the slip meter. The average difference at all rates for the three belts was only about 1.5 ft. per min.

Errors introduced by slight changes of speed or load, or small inaccuracies in determining the rate of synchronism are estimated to be within the allowable experimental error.

DISCUSSION OF RESULTS

The curves of Figs. 3 to 11 are devoted to belt No. 1, showing its slip characteristics with three pulley combinations, namely, a 24-in. driving a 24-in., a 6-in. driving a 24-in., and a 24-in. driving a 6-in. The other curves, Figs. 12 to 20, are devoted to belts 2 and 3 as indicated. The capacity or horsepower-slip curves were obtained by use of the regular slip meter. Horizontal axes of the slip-distribution curves denote the number of degrees around the pulley measured from the first point of belt and pulley contact.

Creep and the combination of creep and slip are well illustrated by the curves on Figs. 12, 13, and 14 from belt No. 2, on 24-in. pulleys. At low loads on the belt, the slip did not begin until the belt had completed at least two-thirds of its travel around the pulley. When it did begin, the slip increased rapidly through the rest of the arc of contact. As the load was raised, the point of initial slip traveled around the arc backward until it reached the first contact point of the driver pulley at about 17 hp. Up to this point the entire slip on both pulleys had been due to the elasticity of the

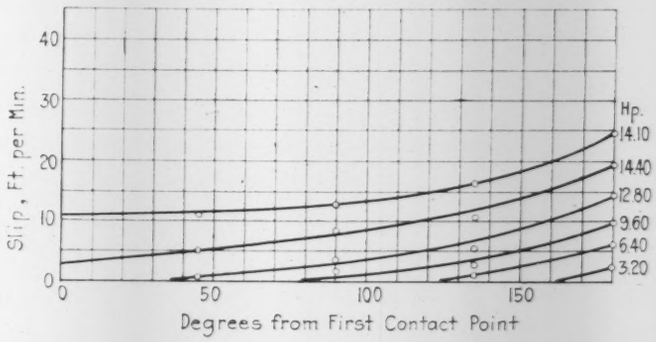

FIG. 16 SLIP DISTRIBUTION—24-IN. DRIVEN PULLEY

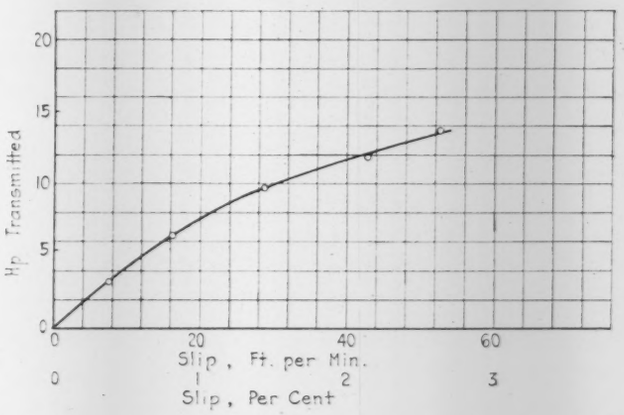

FIG. 18 TOTAL SLIP—BELT NO. 3, 6-IN. PULLEY DRIVING 24-IN. PULLEY

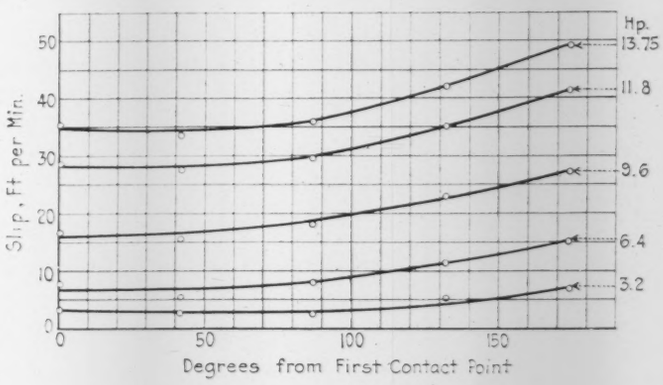

FIG. 20 SLIP DISTRIBUTION—6-IN. DRIVER PULLEY

belting material, and is usually known as creep. At loads above 17 hp., slip was obtained at the first contact point on the driver pulley, indicating the presence of true slippage, which may be considered as any slip not due to elasticity.

The merging point of creep and slip on the driven pulley occurred at slightly over 19 hp. (Fig. 13). Otherwise creep took place in approximately the same way on both pulleys, although the conditions are different on each, because the belt comes on the driver tight and goes off loose, while the reverse is true with the driven pulley. Theoretically the total amount of creep should be the same on each pulley at any given load, because the change in belt length must be the same at either pulley. The data confirm this. For example, at 9.6 hp. the total creep on the driver pulley

was 10.2 ft. per min. (slip at 180 deg.), and that on the driven pulley was 9.8 ft. per min.

An example of combined creep and slip will be found in Figs. 19 and 20. Here the 6-in. pulley drove the 24-in. pulley through belt No. 3, and there was more slip than creep on the smaller pulley. The creep on this pulley was the difference between the slip at the first and last points of contact. At 9.6 hp. the creep on the 6-in. pulley was 27 — 16 = 11 ft. per min., and on the 24-in. pulley it was 11 — 0 = 11 ft. per min., an exact check.

Generally, the accuracy of the data became uncertain when the load carried the belt beyond the merging point of creep and slip. As soon as true slip was reached, the amount fluctuated a good deal, and as all the readings could not be taken simultaneously, it is very probable that the slips did not remain constant while being taken.

Most of the curves of horsepower vs. slip are straight, or nearly so, up to a certain point, after which they bend to the right, and flatten out more or less depending on the belt. In every case, this point closely coincides with the merging of creep and slip, as indicated by the corresponding slip-distribution curves. Figs. 15, 16, and 17 furnish a good example. Up to approximately 13 hp. the slip-distribution curves show creep but no slip, and the curve of horsepower vs. slip on Fig. 15 is straight up to this point. As soon as creep merges into slip at loads above 13 hp. the curve bends over sharply to the right. This merging point can usually be determined quite accurately from any curve of horsepower vs. slip, except for the type of curve shown on Fig. 18, where there is no such point because true slip is present on the small pulley at any load. The break-over point is often gradual, because creep and slip sometimes merge sooner on one pulley than on the other.

Evidently the frictional characteristics of the belt have an important effect on the slip distribution. With belt No. 1 on 24-in. pulleys the creep and slip combined at about 33 hp., while on belt No. 2 this point was reached at 17 hp. On belt No. 3 it was still lower. The merging point seems to be a function of the coefficient of friction, in such a way that an increase in coefficient causes an increase in the merging-point load.

The coefficient of friction of leather increases with the rate of slip. This property also influences the slip distribution. Taking the case of belts No. 1 and No. 2, whose coefficients are known to increase with slip, we find that it was possible to raise the load well beyond the break-over point without causing excessive slip. See Figs. 3 and 12. Beyond the merging point, both creep and true slip increased with the load. In the instance of a belt whose coefficient does not increase to any extent with the slip, such as No. 3 (Fig. 15) the load cannot be raised much above the break-over point, and a slight raise beyond this point greatly increases the true slip without causing an appreciable change in the amount of creep.

The data on the 6-in. and 24-in. pulleys give an idea of the general effect of employing a small and large pulley together. With belt No. 1, creep and slip merged on the 6-in. driver pulley (Fig. 8) at 5 hp., and excessive slip occurred before the limit of true creep had been reached on the 24-in. pulley. In the case of belt No. 3 (Fig. 20), true slip started on the 6-in. pulley as soon as the load was applied. Since true slip was not present on the larger pulley, a majority of the total slip with both belts took place on the smaller pulley. Thus when a small pulley drives a larger one of the same material the merging point is reached sooner on the small pulley, and more slip on this pulley is the result. It is probable that, as the pulleys approach the same size, creep and slip combine at more nearly the same load on each, and the slip becomes more evenly divided. Results from other tests corroborate this conclusion.

When the 6-in. pulley was driven, the slip was less than when it was driving. See Figs. 6, 8, 9, and 10 for belt No. 1. As a driven pulley the merging point occurred at about 24 hp., while as a driver pulley this happened at 5 hp., resulting in more total slip with the latter pulley combination.

The action of belt creep causes an increasing rate of slip as the belt passes around the arc of contact. On the driver pulley the highest rate of slip, and therefore the highest coefficient of friction, is obtained as the belt passes on to the loose side. But with the driven pulley the highest rate of slip coincides with its passage on to the tight side. The distribution of forces between the belt and pulley must be different in each case. This probably accounts for the lesser amount of slip on the 6-in. pulley when driven.

There is need for further research along these lines, such as the determination of the coefficient of friction, and normal pressure of the belt at any point around the pulley. The belting formula $\frac{T_1 - T_c}{T_2 - T_c} = e^{f\theta}$ has been frequently criticized, because f the coefficient of friction is assumed to be constant at all points on the arc of contact, when it is known to vary with the different rates of slip caused by creep. These data combined with a knowledge of the slip distribution and the elastic properties of belting material would furnish the basis of a more accurate formula.

Discussion

C. A. NORMAN[1] wrote that he would have liked to have seen complete references to the work of other investigators. He also wished that the author had stated the magnitude of the zero error.

A. A. Adler[2] discussed the possibility of the author's connecting up his work with theories already published. Something was apparently wrong with the formulas now in use. Thus Kammerer of Germany had driven belts up to 21,000 ft. per min., for which our formulas had shown a cessation of driving power at 9000 ft. per min. The author had evidently not taken into consideration a more modern theory in regard to the lag between stress and strain at very high speed. The time interval in going from the tight side to the slack side was so great that the strain would not correspond to the stress. Inertia of the belt should also be considered. Any theory of belt slip would be incomplete unless that case was taken into consideration where a constant torque was applied on the driving side and an intermittent torque on the receiving side.

C. G. Barton[3] pointed out that in the German experiments at very high speeds, the runs were only of a few minutes' duration. Some years back, in a paper presented before the Society, he had discussed the question of lag.

The author, in closing said that the zero error was really nothing but a zero correction. It was a certain value of slip that was indicated at no load, which was measured and subtracted or added to the readings. It was not an error but simply a correction to allow for the fact that the exact diameters of the wheel could not be obtained in every case. He had made tests up to 10,000 ft. per min. on pulleys of such small diameter as 30 in. and found that power could readily be transmitted at these speeds, but as Mr. Barth had pointed out, a belt would not last very long under those conditions, and that point has been proved pretty well in practice. He had made many tests on the stretching of a belt and releasing it at very rapid rates, in fact, as fast as the belt could be operated at 5000 or 6000 ft. per min., and had found that the lag was practically the same at all speeds within that range.

The author wished to correct the impression of Mr. Adler's that the theory of lag between stress and strain had not been considered. Any appreciable stretch of the leather while passing from pulley to pulley on the tight side, or contraction on the loose side, would have caused an error in the creep measurement. The fact that the readings closely agreed with the total from the regular slip meter indicated that the error from this source was negligible at the belt speed used (2000 ft. per min.). The experiments referred to early in the paper had furnished satisfactory proof of this before these experiments were started. If the belt speed was increased the error from this source would become greater, and it was probable that this method of creep measurement would not be sufficiently accurate at speeds over 6000 ft. per min. At the same time, the conclusions from these experiments could be safely applied to belts operating at all ordinary speeds, say, up to 8000 ft. per min., or more. This included at least 99 per cent of all belts.

The author had not attempted to build up a theory of belt creep and slip, but rather to present experimental results which confirmed, and enlarged the existing theory. Such variables as intermittent receiving torque or inertia would have little or no effect on these results, although they were important in a general treatment of belt transmission.

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[1] Professor of Machine Design, Ohio State University, Columbus, Ohio. Mem. A.S.M.E.

[2] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[3] New Haven, Conn. Life Mem. A.S.M.E.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

Influence of Wetness of Steam on Steam-Turbine Operation

WITHIN the last few years operating pressures of steam turbines have greatly increased, without, however, in many cases, a simultaneous increase in the superheat of the steam. As a result of this practice, particularly where high condenser vacua are used, the expansion of the steam extends deep into the region of saturation, with the further consequence that in the last stages of the turbine the water content of the steam is quite high. Improvements in the efficiency of the steam turbine have also contributed to increasing the wetness of the steam in the last stages, and simultaneously with this, increases in output have brought about increased peripheral velocities, particularly in the same last stages.

The first visible consequence of the operation of these influences appears in the shape of erosion of the blades in the last stages, particularly those of the rotor. Also the efficiency of the low-pressure stages turns out to be below that which was expected. We know that the efficiency of a steam turbine increases with the entrance temperature, and to a much greater extent than the increase in the factor of heat recovery would lead one to expect. This is due to the fact that the wetness of the steam in the last rows of blades decreases with an increase of the entrance temperature. [The words "entrance temperature" have apparently been omitted in the original article through a misprint.—EDITOR.]

BRAKE ACTION OF THE WATER IN STEAM

Fig. 1 shows the velocity triangle of a single stage in a reaction turbine with equal blade angles for the runner and guide blades. The velocity c_{1W} of the water at the exit from the guide wheel is smaller than the velocity c_{1D} of the steam. If G_D and G_W be respectively the weights of steam and water flowing through the turbine per second, the outputs in kilowatts at the periphery of the wheel are

$$L_{uD} = \frac{G_D}{102\,g} u c_{uD} \quad \text{and} \quad L_{uW} = \frac{G_W}{102\,g} u c_{uW}$$

Since, however,

$$c_{uW} = c_{uD} - 2\,c_{1D}\cos\alpha\left(1 - \frac{c_{1W}}{c_{1D}}\right)$$

and since

$$G_W + G_D = G$$

the following expression is obtained for the output at the periphery of a stage:

$$L_u = L_{uD} + L_{uW} = \frac{G}{102\,g} u c_{uD} - \frac{2G_W}{102\,g} u c_{1D}\cos\alpha\left(1 - \frac{c_{1W}}{c_{1D}}\right) \quad \ldots\ldots [1]$$

The first member represents the output which would have been obtained with the water at the same velocity as the steam. The second member therefore represents the brake action L_{W1} of water in a single stage. By using the notation $G_W = (1 - x)G$, the following formula is obtained:

$$L_{W1} = \frac{2G\,(1 - x)}{102\,g} u c_{1D}\cos\alpha\left(1 - \frac{c_{1W}}{c_{1D}}\right)$$

From this the following formula is obtained for all the stages within the region of wet steam:

$$L_W = \frac{2G}{102\,g}\sum (1 - x)\, u c_{1D}\cos\alpha\left(1 - \frac{c_{1W}}{c_{1D}}\right) \quad \ldots\ldots [2]$$

If we introduce the Parsons coefficient

$$X = \frac{zu^2}{H_0} \quad \ldots\ldots [3]$$

we can write with a high degree of approximation:[1]

$$2c_{1D}\cos\alpha = u\left(1 + \frac{g}{AX}\right) \quad \ldots\ldots [4]$$

and thus obtain from [2]

$$L_W = \frac{G}{102\,g}\sum (1 - x)u^2\left(1 - \frac{c_{1W}}{c_{1D}}\right)\left(1 + \frac{g}{AX}\right) \quad \ldots\ldots [5]$$

In this equation all the values up to the expression c_{1W}/c_{1D} are known. Since the velocity of motion of the water cannot be determined by measurement, a basis for estimating its value has to be obtained by the following reasoning.

SIZE AND VELOCITY OF WATER DROPS

It is obvious that the water will be carried by the steam with greater efficiency, the smaller the size of the drops. On the other

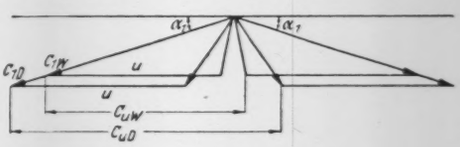

FIG. 1 VELOCITY TRIANGLE OF A SINGLE STAGE OF A HIGH-PRESSURE TURBINE

hand, however, the size of the drops depends on the density and velocity of the steam. Should we introduce a number of water drops into a stream of steam, these drops will be broken up into very small droplets, providing the velocity of the steam be great enough. This takes place whenever the pressure exerted by the flowing steam on the drop is great enough to overcome the capillary forces acting within the latter.

The capillary surface tension σ may be expressed as a function of temperature,[2] namely,

$$\sigma = 7.66 \times 10^{-3} - 1.45 \times 10^{-5} t \quad \ldots\ldots [6]$$

Because of this surface tension there is within a drop of diameter d a pressure

$$p^* = \frac{4c}{d} \text{ kg. per sq. m.}$$

If this drop happens to fall within a stream of steam having a relative velocity W, the force exerted by the steam on the drop is equal to

$$P = \psi \rho_D F W^2$$

Here ρ_D is the density of the steam, $F = \pi d^2/4$, the surface area of the drop, and ψ is a dimensionless function of the Reynolds number $R = Wd/\nu$. In Fig. 2 ψ is plotted (on the basis of Göttingen measurements) as a function of R. If this force is uniformly distributed over the surface of the drop, a pressure is created equal to $\overline{p} = \psi \rho_D W^2$. It may be now assumed that a drop of water is torn apart when $\xi = \overline{p}/p^*$ exceeds a certain value. To test this out an experiment was made with air. Into a stream of

[1] Cp. Stodola, Dampf- und Gas-Turbinen, 5th edition, p. 245. In Eq. [31] members with ξ may be neglected, and then Eq. [27] gives Eq. [4] as shown.

[2] Cp. Stodola, Dampf- und Gas-Turbinen, 5th edition, p. 854.

air flowing with a velocity W, a water drop was allowed to fall, which was broken up by the air current and dashed against a board covered with blotting paper. By measuring the size of the wetted spots, it became possible to determine the diameter of the drops—in this instance by calibration against drops of known size. In Fig. 3 the size of the drops so obtained is logarithmically plotted as a function of the air velocity, the corresponding values of ψ being also given. These tests—as to which no special claim in regard to precision is made—have led to the formulation of the law $d = k\ W^2$, where k is equal to 1.1. It follows from this that

$$\xi = \frac{\psi \rho_D W^2 d}{4\sigma} = \frac{k \psi \rho_D}{4\sigma}$$

In these tests ρ_D, σ and ψ were approximately constant and hence ξ is also constant. By inserting the values of the former we obtain $\xi = 0.87$, which means that a drop will not be torn apart so long as the average externally acting pressure is less than 0.87 times the internal pressure produced by the action of capillary forces. In order to compute the maximum size of such a drop, we may write $\overline{p} = 0.87\ p^*$, which after insertion of the values $\overline{p}$ and p^* becomes

$$\psi R = \frac{3.5\sigma}{\mu W}$$

where $\mu = \nu\rho$ is the viscosity. In Fig. 4 ψR and ψ are plotted as functions of B within the region here considered. This permits determining for a given value of p the temperature as well as σ and μ within the saturated-steam region. Within this region[3] μ is determined by the following equation:

$$\mu = 0.92 \times 10^{-6}\ (1 + 0.00455\ t) \quad \ldots\ldots [11]$$

This permits the computation of ψR as a function of p and W and to the reading from Fig. 4 of the corresponding value of R, which in its turn permits determining the diameter of drop d. In Fig. 5 are plotted the maximum diameters of drops as functions of pressure for the various prevailing velocities and hence the corresponding values of the Reynolds number R. These drop diameters are the maximum possible, but smaller drops may be created through impact against the edges of the blades. The size of the drops being known, it becomes possible to calculate their velocity. If we denote by W the velocity of the flow of steam and by w the velocity of motion of the drops, then in accordance with Equation [8], the equation of motion of the drop will be

$$\psi \frac{\pi d^2}{4} \rho_D (W - w)^2 = \frac{4}{3} \frac{\pi d^3}{8} \frac{\gamma_W}{g} \frac{dw}{dt}$$

or

$$\frac{dw}{dt} = \frac{3}{2} \frac{\gamma_D}{\gamma_W} \frac{\psi}{d} (W - w)^2 \quad \ldots\ldots [12]$$

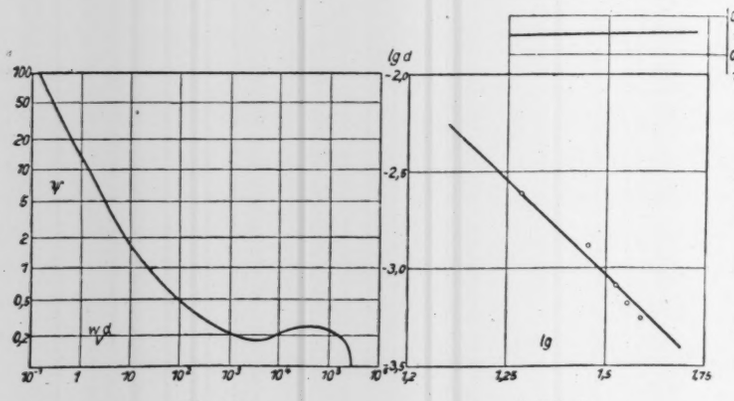

Fig. 2 Fig. 3

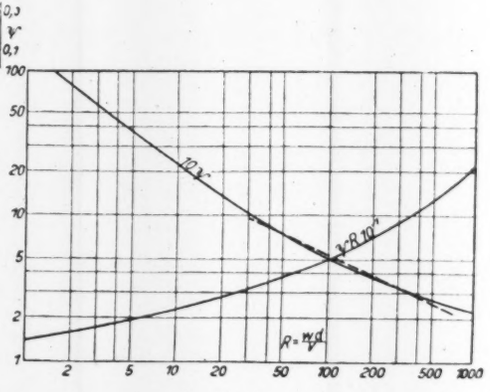

Fig. 4

FIG. 2 COEFFICIENT OF RESISTANCE OF A SPHERE MOVING IN A LIQUID; FIG. 3 SIZE OF DROPS AS A FUNCTION OF VELOCITY OF FLOW OF AIR (DETERMINED BY TEST); FIG. 4 COEFFICIENT OF RESISTANCE OF A SPHERE MOVING IN A LIQUID

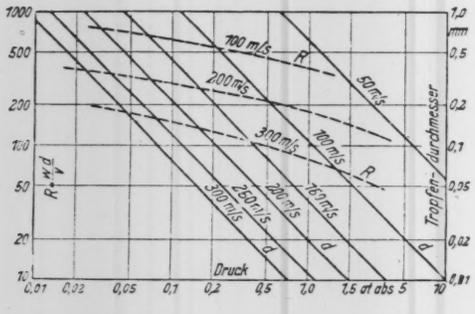

FIG. 5 DIAMETER OF DROP AND CORRESPONDING REYNOLDS NUMBER AS A FUNCTION OF VARYING STEAM PRESSURE AND STEAM VELOCITY
(*Druck* = pressure; *tropfen-durchmesser* = diameter of drop.)

The coefficient ψ as stated above is not a constant but a function of the Reynolds number R. For values of R from 20 to 800, which are possible according to Fig. 5, ψ may be replaced by a simple function of R, such as plotted in Fig. 4, namely,

$$\psi = \frac{5.35}{\sqrt{R}} \quad \ldots\ldots [13]$$

Letting

$$K = \frac{8.0\ \gamma_D \sqrt{\nu}}{\gamma_W a^{3/2}} \quad \ldots\ldots [14]$$

and since

$$dt = \frac{ds}{w}$$

Equation [12][4] becomes

$$\frac{w dw}{(W - w)^{3/2}} = K ds$$

which, integrated, gives

$$\frac{4W - 2w}{\sqrt{W - w}} = Ks + C \quad \ldots\ldots [15]$$

When the drop is picked up by the steam flow, its own velocity is approximately zero. This permits determining the constant C which, letting $w/W = u$, gives

$$\frac{Ks}{2\sqrt{W}} = \left[\frac{2 - u}{\sqrt{1 - u}} - 2\right] = F(u) \quad \ldots\ldots [16]$$

For small values of u it is possible to develop $F(u)$ into a series which, neglecting the higher powers of u, gives

$$F(u) = \frac{u^2}{4}$$

The author next assumes a steam path 2 cm. (0.8 in.) long, over which the water drop is accelerated by the steam up to the point

[3] Cp., e.g., Schüle, Thermodynamik, 3rd edition, vol. 1, p. 334.

[4] In the expression $K = Wd/\gamma$, W is to be taken as the relative velocity $(W - w)$.

where it enters the next row of blades. In Fig. 6 water velocities $w = c_{1W}$ calculated from Equation [16] for various steam velocities are plotted as functions of pressure. From this it appears that at low pressures the water is barely carried over by the steam. One may now calculate the still unknown expression c_{1W}/c_{1D} in Equation [5], emphasizing once more that the size of the drop so computed is the maximum value and hence the corresponding water velocity is a minimum velocity. However, the drops which are formed by condensation in any of the guide or runner blade rows are also extremely small and are carried over with the steam without any trouble. Equation [5] therefore gives somewhat excessively large values for the brake action of the moisture in the steam.

The author proceeds next to give an approximation formula

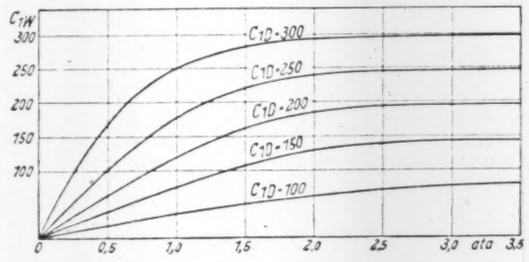

FIG. 6 VELOCITY OF WATER DROPS AT ENTRANCE INTO THE NEXT ROW OF BLADES AS A FUNCTION OF VARYING STEAM PRESSURE AND STEAM VELOCITY

(*Ata* = atmos. abs.)

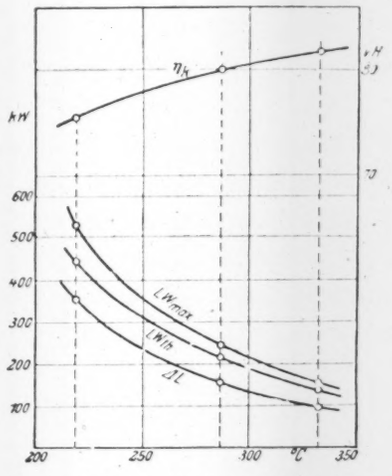

FIG. 7 CLUTCH EFFICIENCY AND BRAKING ACTION OF WATER IN A TWO-CYLINDER TURBINE AS A FUNCTION OF VARYING LIVE-STEAM TEMPERATURES

(%H = per cent.)

for the braking action of water in the case of a multi-stage turbine, and then to test the values obtained experimentally.

TESTS ON A TWO CYLINDER STEAM TURBINE

These tests were carried out in the Steam Turbine Testing Laboratory of Brown, Boveri & Co. at Baden on a two-cylinder steam turbine and at various temperatures of live steam, the steam pressure and vacuum being held constant. The nominal output was 3000 kw. at 3000 r.p.m. and the various values obtained as functions of the live-steam temperature are given in Fig. 7 and Table 1. Of particular significance is the decrease of the measured shaft efficiency η_k found with the decrease of the steam temperature, this being 81.8 per cent at 333 deg. cent. (631.4 deg. fahr.) and 75.6 per cent at 219 deg. cent. (426.2 deg. fahr.), or a decrease of 8.5 per cent.

For every point at which measurements were effected the output was computed which the turbine should have delivered in accordance with known efficiencies of the blading under the existing conditions of various losses and the existing factor of heat recovery. The difference ΔL between the calculated and measured output is said to be due to the brake action of the water. This brake action denoted by $L_{W\,th}$ has been calculated from Equation [5], while the calculated theoretical outputs $L_{W\,max}$ have been computed from the approximation Equation [21]. The ratio $\Delta L/L_{W\,th}$ is on an average 0.73 or about $^3/_4$, and that of $\Delta L/L_{W\,max}$ is on an average 0.64 or about $^2/_3$. From this it would appear that the size of the drops is on an average smaller than has been assumed.

TABLE 1 DATA ON TEST OF 3000-KW. TURBINE

Temperature in front of nozzles, deg. cent.	333	287	219
Pressure in front of nozzles, atmos. abs.	10.87	10.10	10.92
Vacuum, atmos. abs.	0.065	0.061	0.069
Output at the clutch, measured kw.	3470	3155	3180
Output at the clutch, computed kw.	3563	3309	3535
Difference ΔL, kw.	93	154	355
$L_{W\,th}$ (Eq. [5]), kw.	136	216	447
$L_{W\,max}$ (Eq. [21]), kw.	151	246	532
$\frac{\Delta L}{L_{W\,th}}$	0.68	0.71	0.79
$\frac{\Delta L}{L_{W\,max}}$	0.62	0.63	0.67
$\frac{\Delta L}{L_k}$, per cent	2.7	4.9	11.2
Efficiency of turbine at the clutch (η_k) per cent	81.8	80.0	75.6

The simplest way is to assume that the braking action of the water is $^2/_3\ L_{W\,max}$, as this can be computed very quickly.

In an article entitled High Steam Pressure in *Engineering*, vol. 122, p. 636, it is stated that when a stage works in the wet-steam range, 1.03 per cent of efficiency must be deducted for each 1 per cent of wetness of steam. The approximation formula [20] above gives the value of 1.18 for the same condition with $\chi = ^2/_3$ and $X = 3200$. As would appear from Table 1, the braking action of water when the live-steam temperature is 219 deg. cent. is 11.2 per cent of the output at the clutch and only 2.7 per cent for a temperature of 333 deg. cent. These tests show the great influence of wetness of steam in the last stages. There are three ways in which the harmful influence of this condition may be reduced, namely, (1) raise the temperature of the live steam, (2) employ interstage superheating of steam, and (3) remove the water from the turbine. Boilermakers object to an increase in initial temperature of the steam. However, higher temperatures can be employed and, for example, the plant at Langerbrugge only a year ago raised the temperature of steam from 450 to 500 deg. cent. (932 deg. fahr.) with a pressure of 50 atmos. (711 lb.) at the turbine. Because of materials now employed, higher steam temperatures do not seem to be clearly in sight. Interstage superheating of steam is extensively used in America, but does not hold out much promise in the opinion of the author, because of complications which it introduces in plant layout. Undoubtedly the best solution is the removal of water from the last stages and tests now being carried out in this direction would indicate that this can be done in a comparatively simple manner. (Dr. J. v. Freudenreich in *Zeitschrift des Vereines deutscher Ingenieure*, vol. 71, no. 20, May 14, 1927, pp. 664–667, 7 figs, 1 table, *et.*)

Short Abstracts of the Month

AERONAUTICS

The DeHavilland 14-Passenger Hercules Airplane

THE DeHavilland Aircraft Co. has built for the Imperial Airways a special machine to operate in the Near and Far East, commencing on the Cairo-Karrachi route, and which is the first of an order for five machines. The machine is equipped with three Bristol Jupiter engines and will weigh fully loaded 15,100 lb., of which 2900 lb. are given to paying load and crew. The gasoline capacity is 300 British gallons, and the stalling speed for the fully loaded plane is specified at 52 m.p.h.

The machine is controlled laterally by a system of four ailerons. Ample aileron travel combined with the maximum degree of differential effect which the system can conveniently carry without a tendency to take charge at low speed is said to give a most effective control. The main structure of the fuselage is built up of mild-steel tubes with machined steel or duralumin sockets and mild-steel joint plates braced with standard swaged rods. A drawing in the original article shows the details of the main carriage axle. A fuselage braced with swaged rods is said to be more flexible than

one constructed of three-ply wood. For this reason the rear section has been given ample depth and width to withstand the very heavy torsional strains which are imposed upon it, especially when taxiing.

In the performance tests particular attention was paid to the ability to fly with any one or two engines cut out, and also to stability characteristics with various distributions of useful load. The original article gives performance curves of the plane with full load, from which it would appear that it is capable of flying on any two engines with a definite margin of power in hand. (*The Automobile Engineer*, vol. 17, no. 228, May, 1927, pp. 158–164, 11 figs., *dA*)

The German Government and the Junkers Aircraft Company

Statements have been made in the press at various times concerning the relations between the German Government and the Junkers Company. It does not appear that all the facts are available even now or possibly ever will be, in view of which the following information may be of interest.

Just when the German Government became interested in the affairs of the Junkers Company is not clear, but it is known that it came to the company's rescue somewhere about the middle of 1925. At that time the affairs of the company were in a rather involved condition and its indebtedness was 12,000,000 marks as against a share capital of 3,500,000 marks. The company claimed that the Government was responsible for this condition of affairs, the deficit having been caused by the failure to receive payment for goods sold to Russia, in a deal which the Reichswehr promoted. Apparently the Government recognized its responsibility in this case, as it bought 7,000,000 marks worth of preferred shares and gave the Junkers Company a subsidy of 10,000,000 marks. It also subscribed for 80 per cent of the 2,000,000 marks capital of the Junkers Air Navigation Company. Notwithstanding this assistance the company's business went from bad to worse, the number of employees having been reduced from 5000 in 1925 to 2000 in April, 1926, while the indebtedness of the company went up to 18,000,000 gold marks. When, however, the Junkers Company attempted to reduce its working force to 600, the Government objected and the matter was brought before the Federal Supreme Court. Referees were appointed to make a report, which proved to be largely in favor of the Junkers Company and resulted in establishment of much more cordial relations between the parties.

The final result was that the Federal Government gave up its rights to the 7,000,000 marks of preferred stock and cancelled 90 per cent of its claims, these two concessions amounting to a sum estimated at 30,000,000 gold marks. It would therefore appear that since 1925 the Federal Government has contributed to the Junkers Company a sum not less than 40,000,000 marks, and that gives reason to believe that other subsidies were granted to the company before 1925.

The French journal from which these figures are taken tries to explain the behavior of the German Government in this instance as due to its desire to keep going a concern, the existence of which is of such great importance to the German aeronautical industry, but remarks that it would appear that there were other reasons as well, the course of events being such that the Government was in fear of scandalous revelations on the part of the Junkers Company. (*L'Aerophile*, vol. 35, no. 7/8, April 11/15, 1927, p. 101, *g*)

CORROSION

Second Experimental Report to the Atmospheric Corrosion Research Committee

The Committee was established primarily by the British Non-Ferrous Metals Research Association together with the Faraday Society. The first experimental report was presented on December 17, 1923. In the work recorded in the present report the experiments have been considered chiefly with the mechanism involved in the formation of various films on exposure air, with particular reference to the conditions under which protective oxide films may be obtained. The report deals primarily with copper, but data on the behavior of other metals (zinc, brass, aluminum, lead, and iron) are also available. All the data are grouped into two heads—indoor exposure tests and open-air exposure tests, the work being done in the former. (Dr. W. H. J. Vernon in *Transactions of the Faraday Society*, vol. 23, no. 74, April, 1927, pp. 113–183, 29 figs., and discussion pp. 183–204, 3 figs., *eA*)

ENGINEERING MATERIALS

Fatigue of Wood in Alternating Loading

The general impression in the past was that wood is not subject to fatigue such as is induced in metals by alternate loading and unloading or by vibration. This, however, has never been definitely proved experimentally, and the present author claims that his own tests have shaken his conviction in the truth of the above-stated belief.

Tests on small wooden bars bent under static load have shown that the compressed and extended fibers do not behave in the same manner under repeated tests. In investigating causes of weakness of wood in bending as compared with similar wood in tension, he explains the difference by the difference in mechanical properties of the two classes of fibers. The strength of the bar in bending is determined by its fibers under compression and the latter give first by forming folds. In these folds occurs the bending of the harder layers in the resilient medium of the softer layers, which in its turn causes underpenetration of the fibers and a lowering of resistance of the regions in compression under bending as compared with those in tension. The resistance of wood to compression is generally smaller than resistance to tension and the difference between the two resistances is the greater the thinner the hard layers (fall formation, pitchy) and the softer the soft layers (spring formation), this difference, for example, being greater in the case of pine than in that of oak.

If the above is true, it would appear that wood fibers, even under a long-continued compression, will have a lower resistance to rupture than fresh fibers which have never been under compression, and the author describes tests undertaken to check the above statement. From these tests—described in detail in the original article—it would appear that both in individual cases and as an average the resistance to rupture found after preliminary compression proved to be the lowest, from which it would appear that previous compression effects subsequent resistance to tension. It is true that loss of strength induced by previous compression was found to be comparatively small, from 5 to 10 per cent, and the conditions of test were such as to make this loss of strength worthy of attention. Further tests have shown that in alternating loading and unloading tests each cycle appears to affect somewhat the strength of the material, and that a number of such cycles lower materially the resistance of wood to rupture. [First instalment of a serial article in Russian, by Prof. I. Siminski in *Engineering Herald* (*Vestnik Ingeneror*)—no. 4, Apr., 1927, pp. 137–140, 4 figs., *eA*]

Some Experiments on the Effect of Repeated Stresses on Materials

Only the conclusions of this extensive and important paper can be stated here. The conclusion can be drawn that for all practical purposes most materials have a real fatigue range. In the case of the carbon steels and the alloy steels generally used for many parts of airplane engines when heat-treated so as to give varying ultimate stresses, the total fatigue range for equal and opposite stress varies from 0.8 to 1 times the breaking strength, but for other materials it may be less than one-half the breaking stress. Assumptions made as to the behavior of certain types of materials from the results of others without actual experiments may be dangerous.

The range of stress S_r for any material depends upon the mean stress S_m, and for certain steels the range of stress both for bending and torsion can be expressed as $S_r = A - BS_m$, A and B being constants.

The curves of range with variable mean stress for materials such as mild steel more nearly approximate to the Gerber parabola except near to the zero mean stress, when, as indicated by Bairstow, the range remains constant for a certain variation of the mean stress.

When there is a steadily applied stress of one particular kind, as, for example, a torsional stress, the safe range under bending stress

may be unaffected, provided the steadily applied stress does not exceed some specified amount.

The factor of safety when materials are subjected to cyclical stresses should be considered in relationship to the mean stress and the range of stress. This has long been pointed out.

Rapid methods of determining fatigue range, by optical means of thermocouples, in the hands of careful experimenters may be reliable, but they can hardly be taken as convincing evidence without further tests.

Pinholes, sharp corners, and keyways on shafts, rapid changes of shape in metal spars and other elements, particularly those which may be subjected to blows or vibrations, should be carefully avoided. Materials should be as homogeneous as possible. In this connection it would appear that rapid discontinuities, either mechanical or metallurgical, which give opportunities for stress concentration on a finite length in any plane should be avoided; large-grained structure of metals gives opportunities of stress concentration far greater than a small-grained material with crystalline growth well distributed in all directions.

Cold-worked materials like carbon-steel tubes—and brass tubes—have their fatigue ranges increased by heat treatment.

Raising the breaking strength of certain alloy steels by quenching from above the change point and tempering at various temperatures will increase the fatigue range, but the ratio of fatigue range to ultimate stress may diminish.

The key to the failure of materials under repeated stress has not yet been found, but no problem in the materials of engineering is more pressing for solution and can have more interest or economic importance. This paper has been rather rapidly compiled and has only attempted to put forward certain empirical results and theories to lead to a discussion and in the hope that those who are connected with aircraft, in which the problem is of supreme importance, will contribute something to its solution.

In addition to this the following facts appear to have been established: Several points arose out of those early experiments, which have been much more closely followed up by the post-war work. These tests have shown that with certain materials the range of repetition stress did not appear to have any definite relationship with the primitive elastic limit of the material, as then defined, or, to be more accurate, with the primitive limit of proportionality, and that specimens would resist ranges of stress much greater than the apparent proportional range for very many millions of repetitions.

The number of repetitions which would be resisted by a specimen for certain ranges of stress depended upon what might be called the path of loading, or, in other words, the range of stress could be considerably raised by gradually raising the applied stress while the specimen was going through cycles of stress. The author was able to show later that a mild-steel specimen could, by gradual loading, be made to resist for 20 million repetitions, a range of stress 25 per cent greater than the range that would break it in less than 13,000 repetitions, if applied at once to the specimen.

It was also found that even cast alloys could be run for more than 250 million repetitions without fracture provided the range of stress was below a certain amount. Subsequent experiments have shown that very many more than 250 million repetitions can be applied.

Following, as it subsequently transpired, the methods used by Smith, Mason, and Gough, attempts were made to draw load-strain diagrams for specimens subjected to repeated stresses. These soon indicated that for normalized mild steel the load-strain curves showed deviation from proportionality at stresses very close to those which cause fracture at between 50 and 90 million repetitions, but for other materials, cold-drawn steels and brasses, quenched and tempered steels, and alloy steels it was very difficult to determine the fatigue range from such curves. A great deal of work, to be referred to later, has been done upon this point both on the Wöhler type of machine and in the beautiful machine designed by Haigh, but it can hardly be said that except for certain special materials, reliable fatigue ranges can be obtained by this method. (Paper by Prof. F. C. Lea before the Institution of Aeronautical Engineers; abstracted from the *Journal of The Institution of Aeronautical Engineers*, vol. 1, no. 5, May, 1927, pp. 7–41, 26 figs., and discussion, pp. 41–46, *etA*)

FUELS AND FIRING

Economic Limits of Oil Cracking

THE year 1900 may be considered to have marked the turning point between two important eras. Up to that year the refiner's efforts were bent toward the production of the maximum amount of kerosene. In 1900, then, he obtained 58 per cent of kerosene and 9 per cent of gasoline from the total crude oil run. That probably marked the maximum development in this direction.

The industry was only 50 years old, but it had built a very large volume of business with kerosene as its major product. Processes in use at the opening of the present century involved cracking of the heavy residuals at atmospheric pressure. This gave a relatively small amount of gasoline as compared with the production of kerosene.

Already, however, the wider use of electricity and artificial gas for illumination was superseding the oil lamp, and a healthy demand for automobile fuel gave some indication of its future importance. To meet the increasing demand for gasoline, it became necessary for the refiner to take the more volatile fractions from his kerosene.

Take another milestone from which to appraise changes in the industry. By 1914, when Europe went to war, the market's demands had changed to such an extent that production of kerosene was but 24 per cent and that of gasoline 26 per cent of the total crude oil refined. The refiner had come to run his business primarily for supplying motor fuel, in the course of which he had been compelled to increase greatly the amount of crude oil run. A necessary by-product was a large percentage of heavier products for which there was not a ready market. Gas oil was becoming a real burden on the industry.

The situation had been foreseen, and two years earlier the first commercial application of cracking to produce gasoline from gas oil at superatmospheric pressure had been made. Three years later, in 1915, the Standard Oil Company (N. J.) started the fires under its first Burton still. As the demand for gasoline continued to broaden, more and more of these stills were built, until in 1920 the company had in operation 346 Burton stills with a total potential daily production in excess of 1,000,000 gal. of gasoline. From being a burden gas oil had become a desirable product in its new role as a secondary but highly important source of gasoline.

Although the Burton process represented a tremendous advance in the efficiency of refining, it was operated under the serious handicap of requiring a clean distillate product as charging stock. Recognizing this limitation, the Standard Oil Company (N. J.) attacked the problem of developing an improved method which could use the residue from crude oil or even heavy fuel crudes as charging stocks. The reward was the elimination of losses unavoidable in the operation of the original Burton stills.

The tube-and-tank process resulted from this research and development work, and the first commercial units were put into operation in 1920. With subsequent refinements this process has demonstrated its superiority because of its large capacity and its flexibility, which permits the use of any type of distillate, heavy residue, or crude oil.

From 1912, when the first pressure still was put into operation at Whiting, Ind., down to the present the industry has been subjected to alternating periods of oversupply or threatened shortages of crude oil. Nevertheless for ten years following the introduction of the Burton process there was at all times a market for as much gasoline as could be produced, and generally there was a margin between the market price of fuel oil and the finished product, gasoline, sufficient to make it profitable to proceed with further installations of cracking equipment.

A change in this regard began to be noticed with a depression in business in 1921–1922. At the beginning of 1921 there was on hand a reserve supply of 11,000,000 bbl. of motor fuel. By Jan. 1, 1927, this had increased to 39,000,000 bbl. Of late, then, it has been painfully apparent to the industry that it has supplied itself with surplus capacity for the production of motor gasoline.

Other factors than the overproduction of crude oil contributed to this rapid increase. The recovery from casinghead and refinery gases has been trebled, and a big reduction in the loss of light fractions has been brought about by the use of gas-tight tankage all

the way from the lease to the consumer. However, the most important contribution in this regard was the increase in the production of cracked gasoline. At the end of the war this amounted to 10 per cent of our total supply of motor fuel, whereas last year cracked gasoline contributed 30 per cent of the supply.

Like a chess player, the refiner studies out his own moves in relation to the probable changes on the board before him. He went to the expense of building and installing modern cracking units for the purpose of increasing the percentage of gasoline obtainable from a given amount of crude oil. The economic operation of these units rested on a wide enough margin of profit between the wholesale markets for gasoline and the cost of fuel oil or gas oil to justify the expense of running the cracking stills. When either the market for fuel oil strengthened or when because of the increase in the supply of light crude oil of high gasoline content refiners were in a position to make gasoline at a lower cost than it could be obtained from gas oil, the cracking units became temporarily unprofitable.

The situation for some months has been working directly against the further construction of cracking stills, and now it is sound business judgment to continue to operate a large part of them. The country is getting an oversupply of gasoline, and much of it is coming from the increased production of high-gravity crude oils subjected to straight distillation. To run the residual oils through cracking coils would merely add to the daily surplus going into storage. For that reason a great many cracking units have been shut down throughout the country. The Standard Oil Company (N. J.) has shut down 200 pressure stills at its plant at Bayway, 20 at Charleston, and a like number at Baltimore, while the Standard Oil of Louisiana has discontinued running 100 stills, making a total of 340 out of commission. It has been necessary to reduce operating forces correspondingly. No one knows how long the situation as to supply and demand will continue like this. It is much better to store fuel oil and gas oil than crude direct from the well, because in the former case there is no evaporation loss, the more volatile fractions having been taken off in the first distillation.

If the industry is again faced with a deficiency in the current supply of crude oil, putting a premium on fuel oil as a source for a large part of our gasoline, cracking will again take up the slack. (*The Lamp*, vol. 9, no. 6, Apr., 1927, pp. 5–6, *gA*)

HEATING AND VENTILATION

Cooling and Ventilation of the Minneapolis Auditorium

This installation is of interest because of its large size. It is also the first municipal auditorium in which adequate provision has been made for the comfort of its occupants in hot weather. The completed structure, occupying two entire blocks, will measure 564 ft. by 233 ft., and 140 ft. from sidewalk to roof peak. The combined seating capacity is 18,100, the structure being divided into three major units which can be operated independently. The most interesting feature is that involving the cooling equipment, of which only the following information is available now, the rest dealing with the use of effective temperatures in determining the load of a cooling system, not yet being available in print.

Cooling is accomplished by washing the air with cold well water. In this respect Minneapolis is endowed with an asset of no mean consequence. It is underlaid with a water-bearing formation, easy to reach and abundant in quantity. The temperature of this water is 49½ deg. fahr., in the ground. It is pumped from two wells, each 800 ft. deep, with a 20-in. casing down 250 ft. and a 16-in. bore through rock the rest of the way. The level of water in the wells without pumping stands 60 ft. below grade. When pumping their maximum capacity, the level stands 80 ft. below grade, representing a drawdown of only 20 ft., which reflects the abundance of the supply. This water is pumped direct to a series of spray nozzles in each of the air-washing dehumidifiers, reaching them at a temperature of 50 deg. fahr.

This fresh, cold, sparkling water, sprayed in a veritable cloudburst, purifies and cools the recirculated air, absorbing odors and organic matter which otherwise would limit the amount of air which could be recirculated. This water, immediately after being used for air cooling and washing, goes to the sewer. (S. C. Bloom in a paper presented before the Chicago Section of the American Society of Refrigerating Engineers; abstracted through *The Heating and Ventilating Magazine*, vol. 24, no. 5, May, 1927, pp. 53–58, illustrated, *d*)

HYDRAULIC ENGINEERING

Hydraulic-Turbine Tests by the Allen Method

The Allen or salt velocity method of water measurements (The Salt Velocity Method of Water Measurement, C. M. Allen and E. A. Taylor, Trans. A.S.M.E., vol. 45 (1923), p. 285) is based on the fact that salt in solution increases the electrical conductivity of water. A certain amount of salt solution is introduced near the upper end of the conduit and the flow of the solution across one or more pairs of electrodes at other points of the conduit as recorded graphically by an electric recording instrument. From this the discharge in cubic feet per second is found by dividing the known volume of the penstock in cubic feet by the time in seconds. In the fall of 1926 efficiency tests by this method were made on four units for the Rumford Falls Power Co. at Rumford, Me., to determine the discharge horsepower and efficiency of the wheels as well as the overall efficiency of the unit, and calibrate the permanent venturi-meter sections for daily measurement of the discharge of the units. Only certain parts of the article can be abstracted here.

For computing the net head, the elevation of the water column to piezometers at the entrance of the scroll case was added to the velocity head at that point. From this sum the elevation of the tail-water and the velocity head at the exit of the draft tube were subtracted. In computing the overall efficiency, the output of the generator was compared with the theoretical water horsepower as computed from the discharge and the net head. The temperature of the water was 70 deg. fahr. and the weight of water was taken as 62.4 lb. per cu. ft.

For overall efficiencies, the value computed from the test results was transferred to a common gross head of 98.40 ft. in the usual manner. After reduction to a common gross head, curves were drawn through the test points and no derived curves were made. Final efficiencies were plotted against both gate openings and generator output. For the wheel efficiencies, the values computed for the test results were transferred to a common net head of 97 ft. in the usual manner. In transferring to a common head, no correction was made for change in speed. The total range of head during the test was only 8 in. or ⅔ of 1 per cent of the total head.

After reduction to a common net head of 97 ft., curves of brake horsepower and discharge plotted against gate openings were prepared from test data. From these two curves the values for brake horsepower and discharge were used for computing wheel efficiencies. The final efficiencies were plotted against both gate opening and brake horsepower, using the actual gate opening for wheel efficiency and the gate opening indicated by the governor scale for overall efficiency.

At 98.40 ft. gross head the maximum overall efficiency is 89.6 per cent. This value occurred at 69 per cent of the gate opening on the governor scale and at 7090 kw. generator output. The maximum efficiency of the wheel at 97 ft. net head and 200 r.p.m. was 92.8 per cent when delivering 9500 b.hp. at 67.5 per cent of the actual gate opening. This is 85.6 per cent of the maximum power developed at full gate, which under conditions of 97 ft. head and 200 r.p.m. was 11,155 hp. with an efficiency of 84 per cent. (C. M. Allen, Professor of Hydraulic Engineering, Worcester Polytechnic Inst., Worcester, Mass. (Mem. A.S.M.E.), in *Power Plant Engineering*, vol. 31, no. 10, May 15, 1927, pp. 549–551, *e*)

INTERNAL-COMBUSTION ENGINEERING (See Marine Engineering: The Machinery of the "Augustus:" Motor Car Engineering: A Rotary-Valve Automobile Engine)

A Thermodynamic Analysis of Internal-Combustion-Engine Cycles

The principal object of this investigation was to apply the present accurate system of analysis of the cycle to the two leading cy-

cles of the internal-combustion engine and to obtain thereby accurate values for ideal efficiencies under various conditions. The particular feature of the system of analysis is that such factors as dissociation of the carbon dioxide and water-vapor products is taken into consideration as well as the fact that as the temperature falls during expansion, the combustion continues until at the end of the expansion it is practically complete. The temperature at the end of the expansion was also determined.

The two cycles compared here are the Otto and Diesel cycles, certain assumptions (stated in the original paper on pp. 13, 15, and 16) having been made. An equation (No. 44 on p. 27 of the original) has been derived and enables one to calculate the temperature of the mixture of products at the end of the adiabatic expansion.

The following may be mentioned among the results obtained:

1 The efficiency increased with the compression ratio r, that is, the higher the compression the higher the efficiency, other conditions remaining the same.

2 For the same compression, the efficiency increases with the amount of air used. A lean mixture gives a higher efficiency than a rich mixture.

3 The mean effective pressure, and therefore the power, is a maximum when the air supply is somewhat less than 100 per cent of the theoretical amount (Fig. 9 of original). Thus the mixture for maximum power is a mixture of relatively low efficiency.

4 The ideal efficiencies obtained from the various liquid fuels are practically the same. The small differences in the calculated values are without significance in the light of the probable inaccuracies of the assumed specific heats and heats of combustion of these fuels. The conclusion of Tizard and Pye that the ideal efficiency of the motor is independent of the kind of liquid fuel used is verified.

5 The efficiencies of the Diesel cycle, as a group, range higher than the efficiencies of the Otto cycle. However, a comparison of the two efficiencies at the same compression ratio ($r = 8$) shows that the Otto cycle is inherently more efficient than the Diesel cycle. The superior efficiency of the Diesel cycle is due to the high compression ratio that is permitted by the system of operation.

The authors also discuss efficiency standards of internal-combustion engines and give the efficiency basis on the air standard and an expression for the ideal efficiency where n in the air-standard equation is expressed by a somewhat complicated formula. These equations apply specially to octane, but may be used with small error for any ordinary liquid fuel.

The authors proceed next to a theoretical investigation of a more complete expansion cycle, which is one in which the ratio of expansion of the products of combustion exceeds the ratio of compression of the combustible mixture.

The theoretical consideration is supported by experimental data obtained on a Sargent complete-expansion engine, for a description of which see Trans. A.S.M.E., vol. 22, 1901, p. 312. (Prof. Geo. A. Goodenough (Mem. A.S.M.E.) and John B. Baker, Research Asst. in Mechanical Engineering, *Engineering Experiment Station, University of Illinois, Bulletin No. 160*, Jan., 1927, pp. 1-67, 24 figs., *t*)

LUBRICATION

Engine Service Tests of Internal-Combustion-Engine Lubricating Oils Made from California Crude Petroleum

THE present investigation was carried out in the San Francisco Laboratory of the Bureau of Mines. As a general base it was assumed that a lubricating oil for an internal-combustion engine can be considered as giving good service if it lubricates effectively, there is minimum danger of lubrication failure with its use, and its consumption is not excessive. Among other things, it was found that for the oils tested formation of insolubles is seemingly not related to alkali-absorption numbers but is related to oil consumption.

This is of interest, because one oil of each pair tested was nonemulsifying and had a low alkali-absorption number, while the other oil of each pair emulsified readily and at a high alkali-absorption number.

The general conclusion from the tests is that for use in automotive equipment, oils need not be refined to as high a degree as for use in turbines and similar machinery. It indicates that savings may be made in present refining processes and presents possibilities of successfully utilizing crude oils that today are not being refined to make lubricants. Martin J. Gavin and Gustav Wade in report of coöperative investigation by the Bureau of Mines and the American Petroleum Institute. Preface by F. B. Gough, Former Chief Petroleum Engr., Bureau of Mines. *Bureau of Mines Technical Paper*, no. 387, 1926, 57 pp., 1 sheet of graphs, illustr., *ep*)

MARINE ENGINEERING (See Steam Engineering: Modified Woolf-Type Reciprocating Steam Engine on Trawler)

The Machinery of the "Augustus"

THIS is the largest motorship in the world, rated at 32,000 tons, and is being constructed for the General Italian Navigation Co. for service between Italy and South America. She will have accommodation for 2178 passengers and a crew of about 450.

The propelling machinery consists of four M.A.N. and Savoia double-acting two-cycle engines, each six cylinders, 700 mm. (27.6 in.) bore and 1200 mm. (47.2 in.) stroke designed to develop normally 6200 b.hp. at 120 r.p.m. The contract however, calls for an output of 7000 shaft hp. at 125 r.p.m. for 48 hr. continuously.

The engine on test was coupled to a Froude dynamometer and the trials included one of 30 hr. duration with an output of 9000 i.hp. at 125 r.p.m. One of the auxiliary generators of 600 kw. was kept in operation, supplying power for the motors driving the scavenging blower, the oil pump, cooling-water pump, etc.

The fuel consumption was stated to be 138 grams per i.hp-hr., with fuel of 10,000 cal. This equivalent to 0.305 lb. and, as during very comprehensive trials of a similar engine a mechanical efficiency of 80 per cent was attained, including the power needed for the blower motor, it may be taken that this represents a consumption of 0.38 lb. per b.hp-hr.

In the report of the trials it is stated that the regularity of operation, the absence of vibration, and low fuel consumption convinced the technical experts of the suitability of this type of engine for the propulsion of large liners, a statement that is of interest since none of the double-acting engines of this class that has yet been built has a sea speed of over 90 r.p.m.

The three scavenging blowers are installed in the main engine room at a height so that the air can be delivered direct into the scavenging trunk. Two will be in service, with the third as a standby. They are Brown-Boveri machines, each coupled to a 750-b.hp. motor running at 1800–2450 r.p.m., the delivery pressure being from 1.3 to 2.3 lb. per sq. in. The capacity of each blower is from 60,000 to 67,000 cu. ft. per min.

Two oil-fired boilers are available for general services, such as galleys and laundries. The winches for handling baggage and cargo, the warping winches, and windlass are all operated electrically. (*The British Motor Ship*, vol. 8, no. 86, May, 1927, pp. 40–44, illustr., *d*)

MEASURING INSTRUMENTS

A Pyro Radiation Pyrometer

THE field of application of this type of pyrometers of greatest interest to mechanical engineers is in commercial heat-treatment processes. The fundamental principle of this class of radiation pyrometers is the measurement of the total radiant energy (i.e., both heat and light waves) emitted by a hot body, whose temperature is to be ascertained. The theoretical relation between the temperature of a "perfect black body" and the radiant energy emitted by it is given by the Stefan-Boltzmann law, in accordance with which the radiant energy at any particular temperature is a direct ratio of the fourth power of the absolute temperature of a perfectly black body.

In the early radiation pyrometers a concave mirror was used with a thermocouple mounted at the focus of the mirror for receiving the concentrated rays of radiant energy. Dr. Hase introduced the use of an objective lens, and it is to this type that the Pyro instrument belongs. Here the quartz lens of only 45 mm.

(1.77 in.) diameter is used to concentrate sufficient radiant energy to give proper readings. The thermocouple in this instrument consists of an extremely thin wire attached to a thicker supporting wire and a receiving disk fixed over the joint for better receptivity of the radiation (Fig. 8). The whole couple is then mounted in an evacuated glass bulb and adjusted in the axis of the optical system.

The very small mass of the thermocouple results in speedy response to temperature variations and development of higher forces due to the higher actual temperature reached by the thermocouple for a given amount of radiation impinging on it. It is stated that a single reading can be taken by an average pyro-pyrometer in between $2^1/_2$ and 3 seconds. The sensitivity of the instrument is so great that if the standard instrument is pointed directly at the sun or at an electric-arc light, the temperatures cause volatilization of the materials composing the thermocouple. A screen with the central opening is interposed in front of the thermocouple for the purpose of controlling the distance factor and cut-

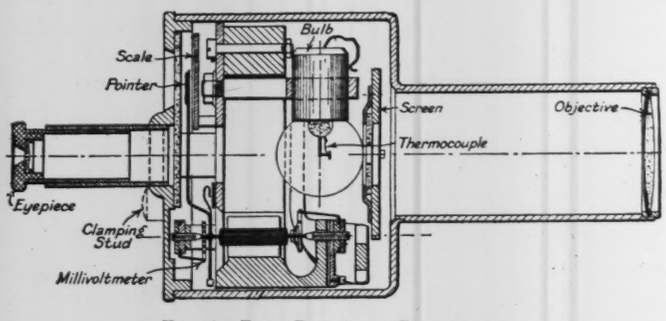

FIG. 8 PYRO RADIATION PYROMETER

ing off side rays radiated from outside the field of vision. The average instrument has a "distance factor" of between 24 and 30, which means that the distance of the instrument from the aperture may be as great as 24 to 30 times the diameter of the aperture. Within the distance-factor limit of the instrument the actual distance between the hot body and the objective lens is immaterial, which makes correct sighting of the instrument comparatively easy.

An interesting feature of the instrument is the incorporation of the millivoltmeter thermocouple and optical system in a single unit, eliminating all accessories such as cable connections and focusing adjustment. The only manipulation required by the operator consists of declamping the pointer by a pressure of the thumb. (*Journal of Scientific Instruments*, vol. 3, no. 4, Jan., 1926, 4 figs., *d*)

rotational speed is a valuable feature, for, while giving almost unlimited port area, it does not impose any limit as occurs in other types by reason of the seating of the valves or the reciprocating motion of the sleeves. At an engine speed of 4000 r.p.m. the valve rotates at practically 666 r.p.m., which is a very low speed in internal-combustion-engine practice.

As the valve is carried in a special type of elastic bearing, it is impossible for it to seize. The diameter of the bore along the head of the cylinder is greater than that of the valve, this difference being

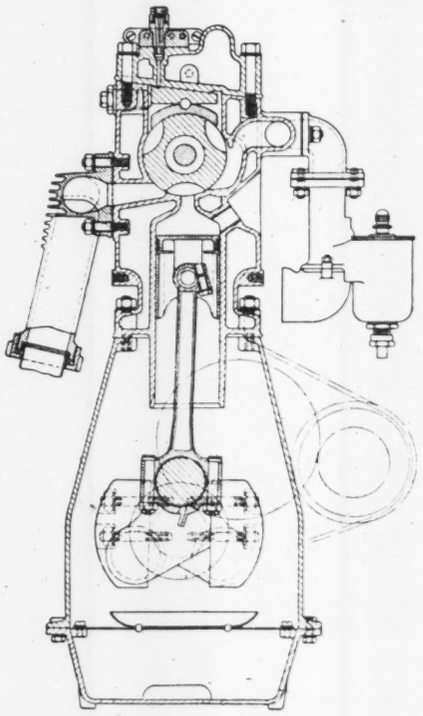

FIG. 9 MINERVA-BOURNONVILLE ENGINE SHOWING VALVE

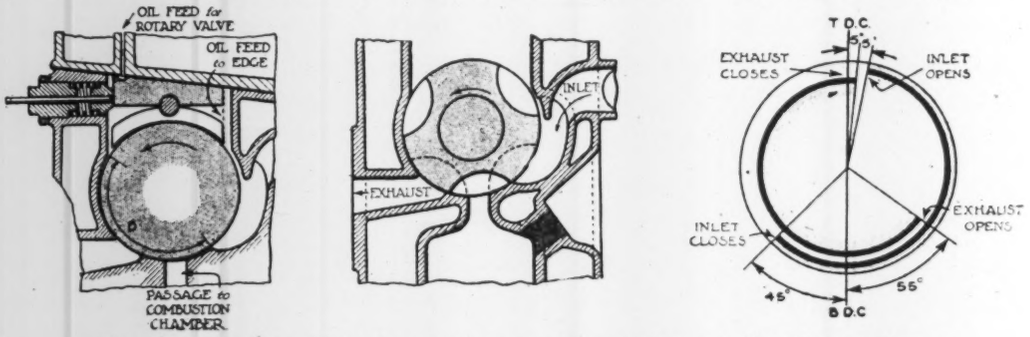

FIG. 10 DIAGRAMMATIC ILLUSTRATIONS OF THE MINERVA-BOURNONVILLE ROTARY VALVE
(Left and center, sequence of operation of the valve; right timing diagram.)

MOTOR-CAR ENGINEERING

A Rotary-Valve Automobile Engine

THE Minerva Company of Antwerp has built a rotary-valve engine known as the Minerva-Bournonville, invented by Mr. Bournonville, a Belgian engineer residing in New York. It may be of interest to mention that the Minerva Company was the first continental firm to produce Knight engines exclusively, so that when it took up the development of the rotary-valve engine it had back of it its long experience with sleeve-valve engines. The new engine, Fig. 9, has a rotary valve mounted on the head of the cylinder with pockets cut on its face by means of which communication is made between the cylinder and the intake and the exhaust passages. For each cylinder there are three pockets on the valve, so that the valve rotates at only one-sixth crankshaft speed.

The valve rotates in the opposite direction to the main shaft, and the pocket which has assured communication with the exhaust port immediately afterward provides for the admission of a fresh charge and thereby gets the benefit of the cool gases. The low

calculated so that under its maximum expansion the valve will turn in the barrel. The barrel is not completely circular, the upper portion having vertical instead of cylindrical walls, and in this is placed a semicircular cast-iron shoe surmounted by a cast-iron wedge. Between the shoe and the wedge there is a recessed steel ball; the upper surface of the wedge is in contact with the diagonal head of the cylinder, and by means of spring-operated studs placed horizontally in the cylinder casting pressure is exerted on the wedge, and through it on the shoe and the rotary valve.

These studs are adjustable through calibrated springs to allow sufficient pressure to be exerted on the valve to prevent its lifting under the explosion. After the engine has been started up the studs are screwed in until there is no movement of the wedge. This movement can be detected with the greatest accuracy by means of a rod going through the center of the stud and touching the face of the wedge. When correct adjustment has been attained, movement of the wedge can take place only for one of two reasons: dust or other foreign matter finding its way between the roller and shoe, or increased friction by reason of temporary lack of lubrication.

In each case the roller would tend to carry the shoe around with it, and by reason of this movement it would cause the wedge to move outward, against the compression of the spring, thus automatically relieving the pressure on the roller and preventing seizure. The steel ball interposed between roller and shoe is not normally under pressure, and its only purpose is to transmit any slight rotational movement of the shoe to the wedge.

The actual arrangement is somewhat more complicated than described above. The lubrication problem has also been solved. The experimental work was carried out on two types of engines: a 4-cylinder model of 80 × 112 mm. (3.15 in. × 4.41 in.) having big ports and the high-compression ratio of 5.91 to 1, and a 6-cylinder type with a compression ratio of 5.2. The former developed 60 hp. at 4000 r.p.m. and the latter 34 hp. at 3000 r.p.m. Several other road tests under trying conditions were made, and it is stated by the Minerva engineer that it is practically impossible to make the valve seize even under the most clumsy handling, there being no reciprocating valve parts. The engine is extremely silent and is expected to maintain its silence almost indefinitely. (*The Autocar*, vol. 58, no. 1646, May 20, 1927, pp. 855–857, illustrated, *d*. Also *The Automobile Engineer*, vol. 17, no. 228, May, 1927, pp. 184–186, illustrated)

POWER-PLANT ENGINEERING

Pulverized-Coal Firing

AMONG other things, the author describes the protected-wall type of combustion chamber. This construction was largely developed by Dr. Lulofs of the Amsterdam Power Station; a somewhat similar type is known as the Bailey wall. In both types the face of the tubes nearest the combustion chamber is protected by refractory tiles which decrease in thickness until thermal stability is attained. The firebrick tiles therefore form a continuous wall, reducing any tendency to produce smoke by chilling the fringes of the flames, and act as a natural heat reservoir assisting the boiler performance.

It is unusual to water-cool the front wall, this being air-cooled only, and in its incandescent condition it assists considerably in accelerating combustion. These walls form part of the boiler heating surface and contribute in a large measure to the boiler evaporation, adding in favorable instances 40 per cent to the output of the boiler. (R. Jackson in a paper before the Institution of Engineers and Shipbuilders in Scotland, abstracted through *Combustion*, vol. 16, no. 5, May, 1927, pp. 279–280, *gp*)

The Benson Boiler Power Plant

A DEMONSTRATION unit with a Benson boiler was installed in 1924 by the Siemens-Schuckert Co. in Germany and has been operated for a total of more than 2000 hours.

The Benson boiler in its general features has already been described in MECHANICAL ENGINEERING (vol. 46, no. 5, May, 1924, p. 288, and vol. 47, no. 5, May, 1925, p. 358) and only a few words in reminder will suffice. The fundamental idea of this boiler is to heat in the upper part of the tube water at the critical pressure of 224.2 atmos. (3295.74 lb. per sq. in.) to a temperature of 374 deg. cent. (705 deg. fahr.). At this temperature and pressure water passes into steam without change of volume and hence without latent heat. As there can be no bubbling of steam under these conditions, there is really no "boiling" in the boiler.

The first unit of practical size was erected in 1922 in England (at Rugby), but as it was capable of producing only 4500 kg. (9900 lb.) of steam per hour, no special prime mover could be provided and the steam was used only to drive a DeLaval turbine. However, the general results obtained were sufficiently encouraging to warrant further work, which was undertaken in Germany by the Siemens-Schuckert Works, because of their interest in the Benson patents.

The Demonstration Unit. In this case the designs were made for a unit to produce 10,000 kg. (22,000 lb.) of steam at 100 atmos. pressure (1470 lb. per sq. in.) and 400 deg. cent. (752 deg. fahr.) at the turbine inlet. The boiler was designed and built by the Siemens-Schuckert Works and the high-pressure turbine by Escher Wyss & Co. Because of lack of space one of the boilers already in existence was used as a feedwater preheater for the projected high-pressure steam generator. This also offered an opportunity to determine to what extent it would be feasible to convert existing boiler plants into the new type. It may be stated at the outset that in so far as present experiments would indicate, this would be economically profitable only under exceptionally favorable conditions.

A Borsig inclined-tube water-header boiler of 13 atmos. (191.1 lb. per sq. in.) operating pressure with a heating surface of 305 sq. m. (3282 sq. ft.) was provided with a higher setting, and beneath it the piping system of the Benson boiler was built in to the first pass (Figs. 11, 12, 13). For this particular experiment oil firing was selected because it was desired to have a method of firing easily adjustable and suitable for the limited space available, the cost of oil having been considered unimportant.

Preliminary calculations indicated that in the raised boiler, water injected cold could be preheated to 180 deg. cent. (356 deg. fahr.).

Because of the subdivision of the high-pressure heating surface the loss of radiation from the flames had to be taken into consideration, and this led to the decision to break the pure counterflow. The boiler tubes *a* (radiation tubes, located next to the combustion space) were supplied from the high-pressure feed pump with water preheated to 180 deg. cent., the pressure being 230 atmos. (3381 lb. per sq. in.). In these pipes, which comprise one-fifth of the high-pressure section, the water was preheated (or superheated as steam) to from 370 to 380 deg. cent. (698 to 716 deg. fahr.). This highly heated water or somewhat superheated steam with a pressure of 225 atmos. (3307.5 lb. per sq. in.) was conducted through bypass tubes to the end of the high-pressure boiler and made to flow in counter-current to the flow of combustion gases through the second part *b*, comprising three-fifths of the high-pressure boiler. Here the steam was superheated to 400 deg. cent. (752 deg. fahr.), then collected and reduced through a throttle valve to 105 atmos. (1543.5 lb. per sq. in.). The last one-fifth of the high-pressure boiler is employed to superheat to about 315 deg. cent. (599 deg. fahr.) the steam previously expanded to 105 atmos., and this part *c*, called for simplicity's sake the superheater, is located between the radiation tubes *a* and the 225-atmos. pressure tubes *b*, which latter are called heaters, as they combine the functions of water heating and superheating. The total surface of the heating elements of the high-pressure boilers (*a*, *b*, and *c*) comprises 216 sq. m. (2323 sq. ft.). The Benson boiler has no steam or water drums. Evaporation, superheating, etc., all take place in tubes of 20 mm. (0.78 in.) inside and 32 mm. (1.259 in.) outside diameter. The tubes are made of specially selected plain carbon steel, it having been found that the use of chrome or nickel alloy steels is unnecessary. The tubes are joined by autogenous lap welding, instead of which, however, either electric butt welding or thermit welding might be used.

To take care of the varying rates of water and steam flow through the different parts of the system, part *a* (water at 230 atmos. = 3381 lb. per sq. in.) has three tubes, while the 230-atmos. steam part has nine tubes, both sets with the tubes connected in parallel.

The reinforcement and method of connecting the tubes are described in some detail in the original article. The flange joints were at first made in accordance with standards for hydraulic piping, i.e., fastened to the tube ends and subsequently welded. Troubles with tubes have, however, developed and this led to employment of the flange joint shown in Fig. 14. It is not so much the pressure as the temperature that causes trouble with packing in such joints, and in the present design these troubles have been in part eliminated by the use of longer bolts. Furthermore, to take care of the larger temperature variations a spring washer is placed under the nut, this washer being one that retains its full elasticity at about 450 deg. cent. (842 deg. fahr.). It is claimed that with this arrangement, notable for its simplicity, they succeeded in producing tight joints notwithstanding all longitudinal variations of the materials. The same joints were used for the steam piping to the turbine.

Equipment. The valves on the feed line and steam exit were made of steel castings with the cone and packing rings of special rustless steel. The water level in the boiler cannot be determined in this case by means of the usual instruments, as the point of transformation of water to steam in the pipes is variable and depends on the load on the boiler. And, furthermore, both water and steam

may flow through the pipe, this producing a certain drop of pressure not less than 1 atmos. (14.7 lb. per sq. in.). The usual water-level indicator was replaced in this case by a venturi meter inserted

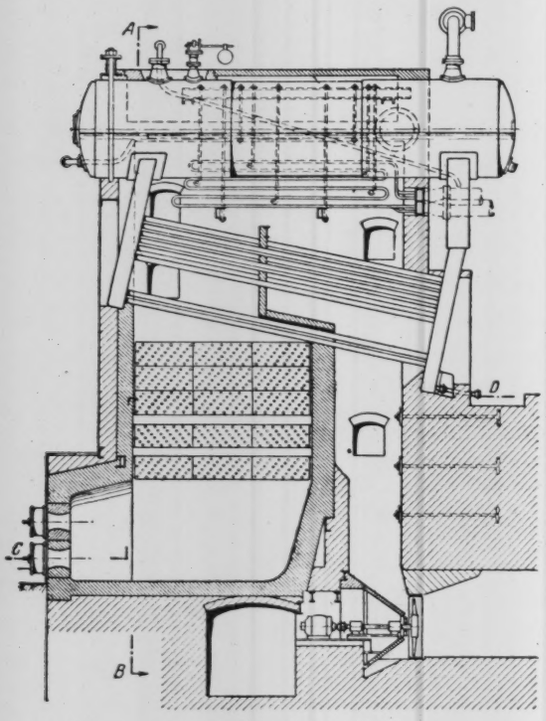

Fig. 11

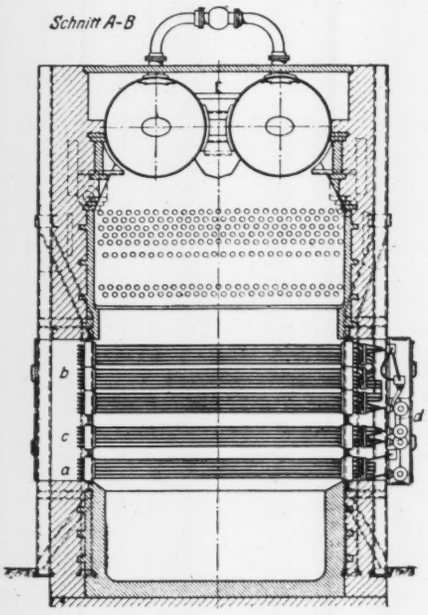

Fig. 12

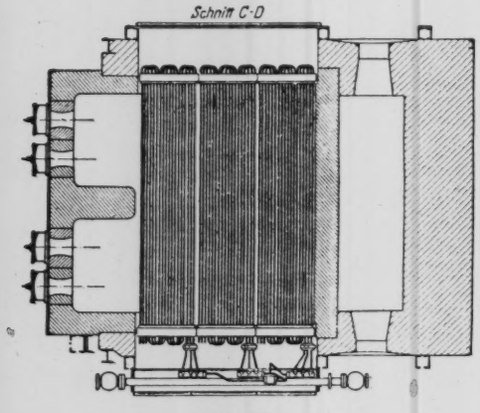

Fig. 13

FIGS. 11 TO 13: EXPERIMENTAL BENSON BOILER INSTALLATION, SHOWING HOW THE INCLINED-TUBE BOILER WAS RAISED TO PROVIDE ROOM FOR THE NEW UNIT

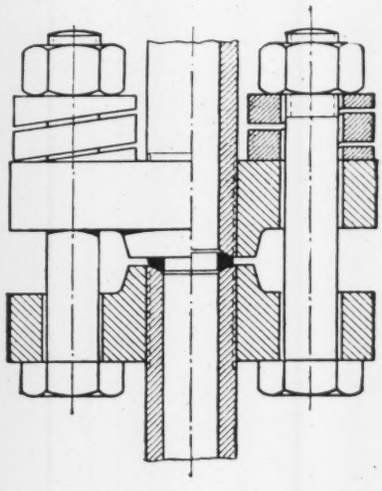

FIG. 14 FLANGED JOINT FOR HIGH-PRESSURE STEAM PIPING (SIEMENS-SCHUCKERT CO.)

in the high-pressure feed pipe. In order for the critical pressure and temperature to be maintained at all times in the meter (parts *a* and *b*) it is very necessary that the supply of water and fuel be properly and jointly regulated. To do this at point *D* (*d* in the original article) Fig. 6, overflow valves—one hand-operated and the other automatic—are provided. The hand valve is an angle valve made of an ordinary steel casting with cone and seat of Krupp rustless steel. The other valve is automatically controlled by an electric transmission and is also of the angle type. Two safety valves are also provided.

The high-pressure pump shown in the original article in an illustration which cannot be reproduced here, is a vertical three-cylinder pump of a type frequently used for hydraulic power transmission. Each of the three pistons is capable of increasing the pressure of the water which enters into the pump at a pressure of 10 to 12 atmos. to about 235 atmos. (from 147 to 3454 lb. per sq. in.) and at full load has an output of 10,000 kg. (22,000 lb.) per hr. The pump is driven by a belt from a direct-current motor capable of speed variation within wide limits, and the power consumption of the pump at full load is 85 kw. As the pump has to handle water at a temperature of 356 to 374 deg. fahr., certain special provisions had to be adopted. The cylinders are installed singly and the gland packing is made of an antimony-lead alloy. The valves and valve seats were first made of bronze, but had to be replaced by better wear-resisting units made of Krupp rustless steel. In handling hot water under the conditions of operation in the present instance, i.e., pressure under about 12 atmos. (176.4 lb. per sq. in.) no hydraulic difficulties were encountered, but at a speed in excess of 140 r.p.m. the volumetric efficiency of the pump fell off somewhat.

It is highly advisable under any conditions to feed a Benson boiler with pure condensate. Table 1 gives the results of tests made.

Experimental Results. During a period of nearly a year and a half of experimentation with the Benson boiler the small water content of the boiler has given no trouble. The boiler has proved to be thoroughly flexible in operation and can, for example, be brought in a few minutes from half-load to full load without any variation in temperatures. When warm water is available the steaming-up period is only from 10 to 12 min. and a boiler can be cut out instantly without having the pipe-wall temperatures run up. As a result the heat losses in bringing the boiler to steaming and banking it can be made extremely small.

Furthermore, because of the very small content of water and steam in the boiler, the latter is endowed with a high factor of safety not only in the sense of boiler laws but actually. Calculations have shown that should a pipe break the resulting increase in pressure would be only 100 mm. (4 in.) of water, and this has been proved in actual operation. A test was made in which the Benson boiler was run on cold water after a permutit treatment, but without the intervention of the low-pressure boiler with its desludging of the water. The result was that sludge accumulated to such an extent in one of the pipes that the latter was completely clogged and ultimately burst. From the outside the bursting of this tube was made apparent only by a hissing noise. The flame was extinguished without causing a back fire in the burners; steam and water appeared at the smokestack, but outside of a small bending of the pipe at the place of rupture, no other damage seems to have been done. The boiler was run the very next day after one of the tube bundles had been replaced by a new one. From the damaged bundle a piece of tube about 1 m. (3.25 ft.) long was cut out and a similar piece welded in by the thermit process. The ruptured place about 14 mm. (0.551 in.) long, clearly showed knife-sharp edges, indicating a flow of metal due to the high temperature, but an investigation of the grain structure in the vicinity of the rupture showed no changes in the metal due to overheating, which would indicate that the temperature must have been around 650 deg. cent. (1220 deg. fahr.). The data in Table 1 were taken during

an 8-hr. run and may be considered as representing operating actual conditions of the boiler.

High-Pressure Turbine. The steam from the boiler was used in a turbine operating at 13 atmos. back pressure, which latter was

TABLE 1 RESULTS OF TESTS OF BENSON BOILER

Steam generated, kg. per hr.	3950	5850	7450	9400
Feedwater temperature, deg. cent.	20	22	23	22
High-pressure feed-pump admission, atmos.	4.2	7.8	9.7	9.7
deg. cent.	139	165	176	176
At entrance to Benson boiler, atmos.	230	229	229	230
deg. cent.	140	166	178	178
Temperature in radiation piping part at the end of the three serially located pipe bundles				
1—deg. cent.	273	291	299	295
2—deg. cent.	343	351	354	350
3—deg. cent.	372	371	368	369
Ahead of overflow valve, atmos.	226	221	212	208
deg. cent.	392	392	390	381
Behind the overflow valve, atmos.	102	102	103.5	106
deg. cent.	311	310	311	309
Behind the superheater, atmos.	100	99.5	100	99
deg. cent.	399	408	408	399
Output of high-pressure feedwater pump, kw.	31.7	46.4	60.2	79.6
Fuel-oil consumption (lower heating value 9950 kg-cal. per kg. = 17,910 B.t.u. per lb.), kg. per hr.	503	678	800	980
Number of burners	4	4	5	6
Fuel-oil pressure, atmos.	5.5	9.0	7.5	8.9
Temperature of exhaust gases, deg. cent.	190	213	250	278
Carbon dioxide content, per cent	11	11.5	12	12.2

the steam pressure at the inlet to a condensing turbine in series with the high-pressure turbine. This latter was a 9-stage turbine running at 10,000 r.p.m. and provided with an undivided housing. The turbine was connected through a Maag gear drive to a 1000-kw. direct-current generator running at 3000 r.p.m. Because of the small volume of steam the governor works by simple throttling. After the initial difficulties were overcome the turbine ran in a satisfactory manner at all loads. For packing at the high-pressure end working at 100 atmos., a special comparatively short labyrinth packing was provided.

Present tests have shown that on a unit of a larger size and with such further improvements as have already become clear a heat consumption of about 2900 large calories (11,513 B.t.u.) per kw-hr. will become possible. A layout in the original article shows how the experimental unit was fitted into the existing plant. This is not at all the best possible layout either technically or economically, but was adopted so as to keep the cost of the test as low as possible.

As a general conclusion, it is stated that the results of the tests of the Benson boiler and the 100-atmos. steam turbine were so gratifying that designs have been worked out for a very much larger steam generator of the same type. This generator is now under construction and will be capable of delivering from 25,000 to 30,000 kg. (55,115 to 66,138 lb.) of steam per hr. to the turbine at 180 atmos. pressure (2646 lb. per sq. in.) and 420 deg. cent. (788 deg. fahr.). This installation, which will be used as a power plant for the cable works of the Siemens-Schuckert Co., will be placed in operation some time during 1927.

The construction of the two generators above referred to leads one to believe that Benson steam-generator-type plants can be built at a price not exceeding that of present-day high-pressure boilers (such as the 35-atmos. type). The further advantage of this new process of steam generation is that it permits obtaining steam of any pressure and temperature up to the limits imposed by the materials of construction now available. Where a plant is being reconstructed a Benson boiler will provide steam for low-pressure turbines and at the same time enable the plant to install series turbines capable of operating with the highest pressures. (Wilhelm Abendroth in *Zeitschrift des Vereines deutscher Ingenieure*, vol. 71, no. 20, May 14, 1927, pp. 657–663, 18 figs., *deA*)

RAILROAD ENGINEERING (See Testing and Measurements: Draft-Gear Testing)

Locomotive Steam Desaturators

THE use of steam desaturators appears to offer an undeveloped field for further improvement in locomotive performance. Furthermore, the trend toward larger locomotive boilers and increased boiler capacity is increasing rather than diminishing the need for practical means to prevent carrying a high percentage of moisture over with the steam to the superheater units and even into the cylinders.

The introduction of baffling devices or other means for precipitating moisture from the steam as it is drawn from the steam space to the superheater units presents a difficult problem in locomotive construction, aside from additional weight and expense of such apparatus, for the reason that unless an outside dry pipe is used, the desaturator must be positioned inside of the boiler. This is objectionable unless it is readily accessible for inspection and maintenance.

The Centrifix desaturator which has quite wide use in stationary practice, has recently been applied to locomotives for test on several roads. This type of desaturator requires no important change in the construction of the locomotive, but can be installed in a steam dome of the conventional construction, by means of a pan which fits into the steam-dome opening and rests on top of the throttle opening so as to separate the steam intake from the main steam space below the dome, and by inserting a number of Centrifix units in this plate. This is supplemented by a cylindrical apron supported from the dome opening in the boiler and suspended from this point down to within 3 in. of the top of the flues.

The Centrifix unit consists of a series of oblique blades that impart a violent whirling motion to steam drawn horizontally into the unit. This throws the moisture against the edges of these blades, from which it is precipitated. The effect of the cylindrical apron is to create a steam pool below the units and prevent a surge of water in close proximity to the Centrifix and a consequent overloading of the units. Separate openings for the dry pipes to the auxiliary turrets are cut in the plate supporting the Centrifix units so that all steam drawn from above this plate is subject to this desaturating effect. The capacity of this device can be augmented by inserting additional units of the same size in the plate or by using standard units of larger dimensions. All units are constructed with monel metal blades.

A test of the Centrifix desaturator recently conducted on a Pacific-type locomotive improved the steam quality from 98.7 per cent without the desaturator to 99.6 per cent with the desaturator. This increase in quality of the steam was noticeable to the extent of about 10 deg. increase in superheat temperature.

Furthermore, it was found that in throwing out the water from the steam all impurities were kept from the cylinders and the inside of the Centrifix pan was clean, while the outside had accumulated quite a scale deposit. This clearly indicates that in addition to the direct effect of removing moisture from the steam and consequently improving the superheat, the desaturator lessens the progressive accumulation of scale in the superheater units and thus, in time, exerts a considerably greater influence upon superheat than is reflected in an immediate comparison of temperatures as above given. (Report to the Railway Fuel Association Convention, Chicago, May 10–13, 1927; abstracted through *Railway Age*, vol. 82, no. 25, May 21, 1927; complete report of convention is given on pp. 1508–1519; the part which deals with desaturators is on p. 1514, *dg*)

The German Gotha Internal-Combustion Railway Motor Car

THE whole engine is accommodated in a special frame, which can be easily exchanged, in such a way that no part of it projects into the car body. The floor of the coach goes straight through from end to end, nor is the arrangement of the seats hampered in any way by the engine plant. All parts of the engine are easily accessible from the interior of the car through trap doors in the floor. The drive is effected by a 6-cylinder 76-hp. NAG engine, fitted with an electric starter, a lighting generator, and a cooling-water pump. The drive is transmitted from one end of the engine through a change-speed gear, which is connected to the reversing gear by a hollow shaft; and from the reversing gear, which can be operated by a claw clutch, to a spur-wheel reduction gear combined with it, the large gear wheel being fixed to the axle. From the other end of the engine a direct-coupled air compressor is driven which produces the compressed air required for operating the change-speed gear. This four-speed gear is designed exceedingly short and sturdy. The clutches for the various speeds are of the laminated disk type and are easily accessible after unscrewing the gear-box lid. The connection of the driving shaft between the change-speed gear and the axle drive is made by flexible disk couplings instead of the expensive Cardan joints. These disk couplings allow

the shafts a relatively great angular play without appreciable loss of power.

The cooling water needed for the engine is recooled in a system of ribbed pipes, located on the roof, by utilizing the natural draft created when the coach is in motion. The cooling is intensified by the draft from a fan, driven by the engine, the air duct being taken through the lavatory compartment.

The two-axle coaches are manufactured in series, many of their parts being standardized. Particular attention has been devoted to the series manufacture of the four-axle coaches, which are to have a seating capacity of up to 120 passengers; their total lengths will be 13,000 mm. (43 ft.), 17,000 mm. (56 ft.), and 20,000 mm. (66 ft.), while the widths will be 2100 mm. (7 ft.) for narrow-gage lines, 2500 mm. (8.2 ft.) for local lines, and 3000 mm. (10 ft.) for both main and secondary lines. The frame carrying the engine is suspended near the center pin of the bogie in such a way as to deaden the noise as much as possible. The floor is quite level from end to end. The change-speed gear will be simplified still further, so as to obtain the four speeds with only two shafts and three pairs of gears. The operation of the change-speed gear will be effected with compressed air, likewise the coupling of the engine with the change-speed gear. A new type of M.A.N. solid-injection Diesel engine with an output of 75 to 90 hp. will be installed in cases where crude oil is to be used as fuel. (*Engineering Progress*, vol. 8, no. 4, Apr., 1927, p. 108, 1 fig., *d*)

REFRIGERATION

Comparative Tests of Household Refrigerating Machines

AN EXTENSIVE test having for its purpose to show the comparative results of operation of several different types of household machines, each one operating on several different types of refrigerators. The interest of the test lies in these comparative results, which are presented in the form of numerous curves. It does not appear that the publication of one or two of these would be of particular interest, and there is certainly no space available here to publish enough to give a clear idea of the highly interesting results obtained by the author. Attention of all those interested in the matter is called to the paper itself.

Among the facts established by this investigation, the author calls particular notice to the wide variety of temperature ranges within three different types of regenerators tested as in the same refrigerators with three different types of cooling units of different makes and designs. This temperature variation is the result of several points of fundamental design, such as the size and arrangement of baffles, air ducts, proportion of width to height, door leakage, cooling unit, clearance in the ice chamber, and like points all altering the effective static head of air available for setting up circulation within the refrigerator, and the author cites an example of impeded circulation due to duct and baffle design and the like.

Because of this he considers it impracticable to take one or the "mean" of even two or three temperatures within a refrigerator and call them the true "average" as affecting heat transfer; to take another temperature out in the room somewhere and then try to make accurate use of well-established heat-transfer coefficients to determine or account for actual heat losses. Too many errors enter into these assumed average temperatures. Temperatures immediately adjacent to the ordinary domestic refrigerator will vary from 10 to 15 deg. on the six sides, even with fan circulation of air in the room. Because the determination of the accurate values would require a great deal of additional painstaking work beyond the scope of this paper, all values have been placed on a comparative basis only, even though it was necessary to set up certain quantitative results for the purpose of comparison. The determination of a workable technique for securing accurate "averages" of air temperatures both within and without a domestic refrigerator would be highly desirable and should receive prompt attention and investigation. The author is satisfied that it is a man-sized job, which would require months of study and experiment.

Most installations today are set and operated to freeze ice in trays from one meal time to another without regard to the refrigerator temperature. Very little consideration is given to efficiency.

While the loss due to poor economy may appear small for each individual installation, should the predictions for the near future be fulfilled, which are 1,000,000 machines in operation, the unnecessary motor loss alone will cost the public $20,000,000 per year. (Geo. B. Bright, Cons. Engr., Geo. B. Bright Co., Detroit, Mich., in *Refrigerating Engineering*, vol. 13, no. 11, May, 1927, original article pp. 323–350, discussion pp. 350–352, one table, extensively illustrated, *e*.1)

STEAM ENGINEERING

Modified Woolf-Type Reciprocating Steam Engine on Trawler

A RECIPROCATING steam engine fitted in a large German trawler, the *Claus Bolten*, is of the double compound Woolf type, but differs from similar types in two important details, both of thermodynamic advantage.

In the first place, the steam flowing through the engine is governed by a single piston valve for each of the two identical halves of the engine, and in the second the low-pressure cylinder works on the uniflow principle. From the boiler the steam goes through a valve into the high-pressure cylinder, and after being partially expanded, returns through the valve and some small connecting passages into the low-pressure cylinder, where the expansion is continued and completed. Just before the stroke of the low-pressure is completed, the piston opens ports in the waist of the cylinder wall at half its height, and the steam exhausts through the passage surrounding the waist of the cylinder into the condenser. There is no receiver between the two cylinders, and the two pistons are set at an angle of 180 deg. There remains, however, an excess of low-pressure steam in the low-pressure cylinder, and therefore the valve reopens for a short time while the low-pressure piston is reciprocating, so that the surplus steam may escape to the condenser through an auxiliary steam passage.

As the cooling surfaces of the engine are extremely small, and as the thermal conditions in the low-pressure cylinder are excellent because the steam flows in one direction only, there is a minimum of internal condensation, and the admitted steam is fully used for generating power. In consequence the fuel and steam consumptions are low. The engine of the *Claus Bolten*, which gives 525 i.hp. at 106 r.p.m., consumed in her 6-hr. trial trip under seagoing conditions only 1.15 lb. of good German coal per i.hp-hr., and this result has been fully confirmed during three months of service. The steam, which has a pressure of 200 lb. per sq. in., is superheated, but no forced draft is applied to the boiler.

Experience has proved the engine to be very handy for trawler service, and its economy is shown by the fact that the *Claus Bolten* is able to sail a distance of 8000 to 8400 miles with a bunker capacity of 210 tons and to remain at sea for 34 to 36 days. A second trawler which is under construction will be fitted with an engine of the same kind. It is intended to standardize the engine for powers between 400 hp. and 2000 hp. in eight different sizes. (*Times Trade and Engineering Supplement*, vol. 20, no. 462, May 14, 1927, p. 206, *d*)

TESTING AND MEASUREMENTS

The Beard Cupping Test for Thin Metal Sheets

IT IS CLAIMED that this test will give an indication of the drawing qualities of any thin sheet metal independent of its thickness. In the machine used for the test a uniformly distributed load obtained by oil pressure is employed. The test piece is held firmly clamped in a brass jig and the oil pressure is applied from a small hand-operated pump.

An empirical formula has been obtained for the maximum deflection of a flat circular plate when subjected to any pressure sufficient to produce a certain amount of permanent set. Up to a certain point the curves appear to follow a law of the form $p = ah^n + c$, where p is the pressure and h the depth of cup. The index n and the constants a and c have to be determined, and the original article gives them for copper, brass, and tin plate of certain thicknesses. Tests were also made with different rates of loading to ascertain the effect of time, which was found to have a certain effect but no greater than in the tensile test.

One of the main objects of the author's research was to obtain

a "ductility factor" which would give an indication of the drawing qualities of any thin metal independent of its thickness. After preliminary experiments a standard die, $1^1/_2$ in. in diameter with a $^3/_{16}$ in. radius, was adopted. A high-grade mild steel was selected and portions of each sheet were subjected to various heat treatments after annealing in a closed metal box for an hour at 1000 deg. cent. Cupping tests were then made on each sheet, and the pressure-depth curves plotted. From a cross-plot taken on these curves it appeared that they were of the form—

$$h = a / t^n$$

where h is the depth of cup and t is the thickness of the plate. Further experiments showed that the index n could be taken as constant over a certain pressure range and equal to 0.75, and it was also found that a was constant for a given applied pressure and a given material.

The author therefore writes his "ductility factor" D in the form

$$D = ph\,t^{0.75}$$

where p is the applied pressure in pounds per square inch, and must be the same in any two comparative tests.

For steel sheets of thicknesses 25 to 8 gage, p may be conveniently taken as 1500 lb. per sq. in. It should be noted that the values of n and a may depend on the shape of the die used. Those given in the paper are therefore only strictly applicable to the particular die used for these tests. Tables of ductility factors obtained by the author on sets of specimens are given in the paper, and show good agreement.

Comparisons with the Erichsen method were made by testing similar specimens by both machines. Sheets 0.026 in. in thickness were chosen, and the results obtained are given in the following table:

Material	Erichsen value	Beard's ductility factor
1	10.50	168
2	6.25	123
3	6.15	117
4	5.90	109

The advantages claimed for the author's method may be summarized as follows:

1 The ductility-factor values are as satisfactory for gages of material below 0.07 in. as Brinell hardness numbers for thicker material.

2 The qualities of materials of different thickness can be compared.

3 With the Erichsen test the rate of loading is dependent on the speed at which the operator moves the cupping tool into the test piece. It has been shown in these experiments that time has a considerable influence just before fracture occurs. In the Erichsen test, the quality of the material is determined at the point of fracture and it is difficult to observe the first sign of this, whereas in Beard's test it is determined at a point where time has very little effect.

4 The stretch of the metal is more uniform over the surface of the impression, and takes place over a larger area in those tests in which the cup is produced by forcing a round-nosed tool into the material. (Geoffrey G. Beard in *Journal of the West of Scotland Iron and Steel Institute*, vol. 34, pt. 11, Dec., 1926, pp. 3 and following. Compare *The Metallurgist* (supplement to *The Engineer*), Apr. 29, 1927, pp. 59–60, *de*)

Draft Gear Testing

THE American Railway Association has made arrangements to start at the Engineering Experiment Station of Purdue University tests on all models of freight- and passenger-car draft gears made by the fourteen draft-gear manufacturing companies in America. It is said that railway companies have estimated that they pay annually $50,000,000 in freight-claim damages and $150,000,000 in repair of their rolling stock as a result of cars bumping together too violently. The increased speed at which the railroads are handling their cars in order to improve their services has brought about this increased expenditure, making it necessary to increase the efficiency of present draft gears. The tests will require about two years' time and a special building has been erected to house the testing machine. The latter is of a vertical type with a solid steel block 8 ft. long, 6 ft. wide and $27^1/_2$ in. thick to support the draft gears during the test. Among the tests which the draft gears will undergo is their subjection to the impact of a 27,000-lb. weight from heights up to a maximum of 36 in. as often as five times per minute. Recording instruments of various types will detect the action of the draft gear each time the impact is made. The machine, made by the Tinius Olsen Co., is said to be the largest of its type in existence. (M. M. Boyd in *Purdue Engineering Review*, vol. 22, no. 4, May, 1927, p. 13, 1 fig., *g*)

VARIA

The Institute of Refrigeration at Karlsruhe, Germany

THE Technical High School at Karlsruhe has now an Institute of Refrigeration, of which Plank has been appointed director. Among the equipment is an ammonia machine with an output of 32,000 frigories per hour, consisting of a double-acting horizontal compressor, a counter-current condenser, and an evaporator. Another ammonia machine with an output of 400 frigories per hour has a high-speed vertical compressor, an immersion condenser, and an immersion evaporator, while a third ammonia machine has a rotary compressor driven by an electric motor, and immersion-type condenser and evaporator.

An Audiffren and Singrün sulphurous-anhydride machine of 3000 frigories per hr. with automatic temperature control is also available, together with a carbon-dioxide machine with an output of 5000 frigories and a methyl-chloride machine of smaller output (800 frigories per hr.).

A number of small absorption machines are also available, likewise a unit that will liquefy 3 liters per hr. This latter unit works in connection with one of the ammonia machines which acts as a precooler of the air. The purpose of the Institute is primarily physical and chemical research in matters regarding the production and application of cold.

Among the investigations already under way are measurements of viscosity and heat conductivity of liquefied gases; study of apparatus for measuring the quantity of ammonia in circulation; heat conductivity of snow; and conservation of fruits and vegetables by cold. (*Le Génie Civil*, vol. 90, no. 21, May 21, 1927, pp. 513–514; Compare *Zeitschrift des Vereines deutscher Ingenieure*, vol. 70, no. 51, pp. 1693–1699, Dec. 18, 1926)

Paint for Sealing Joints Against Oil Leaks

A PAINT that effectively seals joints against oil leaks, and also prevents leakage of water and gas, has been produced for the many purposes for which red or white lead, shellac, and a mixture of litharge and glycerine have been used.

The new material, which is dark red in color, requires no priming and can be applied by brushing or dipping. Denatured alcohol is used as a thinner. It dries rapidly and produces a hard, smooth, glossy film which is easily cleaned and which prevents excessive collection of dirt and conducting material, thereby decreasing surface leakage and subsequent carbonization of the surface when used with electrical apparatus.

One of the first fields in which the paint has been applied is in the manufacture of fuel-oil burners, in which one company is now using the material to seal all joints. Other fields in which there will be applications for the paint include plumbing, chemical plants, repair and service shops, packing houses, shipyards, central stations, electric railways, mines, and the manufacture of oil tanks, oil-burning locomotives, and similar equipment. This sealing compound was developed in the Research Laboratory of the General Electric Company and has been designated G-E No. 880 Red Protective Paint. (*General Electric Review*, vol. 30, no. 5, May, 1927, p. 263, *d*)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

White Sulphur Springs Meeting an Unqualified Success

First Spring Meeting of the Society to Be Held at a Resort Since 1910 Highly Successful from Standpoints of Both Technical Sessions and Entertainment Events—Wilfred Lewis Recipient of A.S.M.E. Medal

WHATEVER apprehension may have existed in the minds of members of the Society concerning the outcome of the latest of its Spring Meetings, and the first to be held at a resort since 1910, were dispelled early on the morning of the first day, and the remainder of the time, from all appearances, was devoted to unrestrained participation in the various technical and entertainment events filling the four days May 23 to 26, 1927. Contrary to what is always expected in industrial centers, where there is sure to be a large local delegation, the entire attendance was drawn from outside the immediate district of the meeting. One found members from practically every state east of the Mississippi, and many of the far-western states were represented.

An attempt was made to administer the technical sessions and entertainment events in about equal doses, and there is no doubt that the plan worked out admirably. It was most refreshing to sit through the sessions and listen to the spirited discussion, and then in the afternoon amble out to the golf course and see the opposite side of these staid, calculating engineers and professors as manifested in such events as the golf tournament, or the novelty golf match, where the problem was not hydraulics, fuels, or mathematics, for instance, but how to drive a golf ball out of a stream of water without getting too wet, or figuring out the angle of approach to a barrel so that a ball might be driven through it without a most surprising tendency to spin around the interior. It has been said that "all work and no play makes Jack a dull boy;" those who visited White Sulphur Springs and saw several hundred engineers in both serious and playful moods know that some play and some work, properly mixed, make Jack a most keenly alert boy.

Paralleling the Meeting the Fourteenth Western Meeting of the American Society of Refrigerating Engineers was also held at the Greenbrier, with sessions on Monday morning and evening, and on Tuesday and Wednesday afternoons. This arrangement worked out very satisfactorily, since many of the members of the A.S.R.E. are also members of the A.S.M.E., and the arrangement of alternating morning and afternoon sessions gave them opportunity to enjoy fully both meetings.

Council Meeting

THE Meeting formally opened on Monday morning May 23, at 9:30 with the meeting of the Council. Two sessions were held, with President Schwab presiding at both sessions.

The morning session was devoted almost entirely to discussion of the budget for 1927–28, the single exception being the matter of the next Spring Meeting, which, upon the joint recommendation of the Meetings and Program Committee and the Local Sections Committee, was voted to be held in Pittsburgh, Pa., during April or May, 1928.

The afternoon session convened at 2:30, with the 1927–28 budget again the subject of discussion. After the Special Committee had made its report and those present had been made cognizant of the various needs to be met, approval was voted.

Mrs. Roy V. Wright reported for the Woman's Auxiliary as its Chairman. The members present voiced general approval and appreciation of the splendid work done by this organization.

Other business transacted included the report of the Executive Committee and Presidential Appointments. The Standing Committees of Council also reported at this session, those represented being the Meetings and Program Committee, the Publications Committee, the Membership Committee, the Committee on Professional Divisions, the Local Sections Committee, the Committee on Awards, the Power Test Codes Committee, the Committee on Education and Training for the Industries, and the Standardization Committee.

Technical Committee Meetings and Conferences

AVAILABLE time during the first three days of the meeting was made good use of by a number of Technical Committees in holding meetings. In most cases these gatherings took place following the morning Technical Sessions.

Research Meetings

Main Research Committee. On Monday evening the Main Research Committee met for dinner at 7:00 o'clock, adjourning afterward to the Committee cottage where the meeting was continued through the rest of the evening. The entire committee membership was in attendance.

The appropriation by the Council of $14,000 for Research for the coming fiscal year, October, 1927–1928, was reported. The Main Committee is greatly encouraged in its development of a new and enlarged research program for the Society by the interest and support of President Schwab. It is planned among other things to present this program for discussion at the Open Session on Research to be held at the time of the December, 1927, Annual Meeting.

In line with its policy of stimulating and correlating both its own and other research work in mechanical engineering throughout the country, ways and means of giving these activities greater publicity were discussed. Several new suggestions were made and adopted.

One of the most important and at the same time most difficult matters of business that confronts the Committee at its meetings is the consideration of new research projects. The Committee is constantly appealed to from various quarters to develop and direct important projects for which there is great need or to take on and guide existing projects that need better organization and stimulation. The amount of the annual research appropriation by the Council of course limits the scope of the Committees' activities in this direction, so that many worth-while research projects must be turned down or directed elsewhere.

After consideration of certain matters requiring formal action, the meeting was adjourned due to the lateness of the hour, review of Special Committees' progress being referred to the Chairman and Secretary for report.

A.S.M.E.—A.S.R.E. Conference. Following the recommendations of their respective Councils, representatives of The American Society of Mechanical Engineers and the American Society of Refrigerating Engineers met on Tuesday at 1:00 p.m. at a joint luncheon conference to consider the advisability of undertaking a joint research on the physical constants of several refrigerants. Messrs. H. Harrison, F. G. Hechler, G. A. Horne, and C. C. Spreen, representing the two societies, and Messrs. C. B. LePage and C. G. Worthington, of the A.S.M.E. Staff, were present. Research work along the above-mentioned lines is now being conducted by the Bureau of Standards for the A.S.R.E., but it is felt that joint sponsorship with the A.S.M.E. would make possible a far more comprehensive investigation. It was the opinion of the conference that a joint research committee should be formed to organize and direct the investigation as proposed, and recommendations to this effect have been made to the Main Research Committees of the two Societies.

Fluid Meters Committee. At 11:00 o'clock on Tuesday morning the Special Research Committee on Fluid Meters met in the Main Research Committee's cottage adjoining the Greenbrier Hotel.

Progress of the Sub-Committees working on Parts 1, 2, and 3 of the Fluid Meter Report were reviewed at length. J. M. Spitzglass announced his impending trip to Germany, where he will spend some time on business, and also expects to have an opportunity to

do considerable experimental work for the Fluid Meters Committee. The scope of this work was discussed in detail, including the possibilities of coöperation with similar researches that are now going on abroad.

Mechanical Springs Committee. The meeting of the Special Research Committee on Mechanical Springs began with luncheon on Wednesday and was continued at the Main Committees' Cottage well into the afternoon. The Executive Committee had met the previous day in Washington, D. C.

Encouraging response to the financial campaign now being carried on by the Committee was reported. Beginning about July 1, 1927, a research worker is to be employed full time on the amplification of the Code of Design proposed at the A.S.M.E. Spring Meeting in 1925 by J. K. Wood.

Certain revisions attending the make-up of the Bibliography on Mechanical Springs have delayed its publication, but there is every indication that it will appear within the next two months.

STANDARDS COMMITTEE'S MEETINGS

Milling Cutters. On Tuesday morning the Committee on the Standardization of Milling Cutters held a three-hour session in the living room of one of the cottages.

The Committee first discussed the reports presented by Chairman T. R. Jones of the Sub-Group on Profile Cutters. This report covered proposed standards on Metal Slitting Cutters; Screw-Slotting Cutters; Plain Milling Cutters, Side Milling Cutters, Angular Cutters; Two-Lipped Slotting End Mills.

The report presented by Mr. Jones was verbal and in some detail, since his Sub-Group had held a seven-hour session the previous day during which much of the material previously issued to the Committee in mimeographed form had been somewhat modified and the number of cutters of the various types considerably reduced.

Mr. Jones also reported that the proposed standard on Shell End Mills and Cutter Clamping Screw had been set in page-proof form and had been distributed to the technical press, interested firms, and individuals for criticism and comment.

Erik Oberg, a member of Sub-Group No. 3 on Nomenclature, reported for this Sub-Group and led a detailed discussion on the report which had been mailed in mimeographed form to all the members of the Committee. Some modifications were also recommended and the Secretary was authorized to proceed with the setting of this report in type for further distribution to interested bodies for criticism and comment.

Machine Tapers. The Committee on the Standardization of Machine Tapers held a meeting on Tuesday afternoon. First on the order of business was the election of a permanent Chairman. Ernest F. DuBrul, General Manager, National Machine Tool Builders' Association, who had served as Temporary Chairman, was elected. The Committee then voted to recommend to the sponsor organizations the appointment of additional engineers to round out the representation on the Committee, special attention to be given to the "User Group."

The Committee discussed in a virile way the present practice relative to the manufacture and use of machine tapers commonly known as the Brown and Sharpe Taper, Morse Taper, and Jarno Taper. It was greatly helped in this by the papers which had been previously prepared under the direction of F. O. Hoagland, Master Mechanic, Pratt and Whitney Company, a member of the Committee. These papers included Mr. Hoagland's recommendations for a modification of the Morse Taper.

Sufficient space is not available to record at length the arguments which were advanced for and against the present practice, but it is believed that readers will be interested to know that there is strong sentiment in the Committee for the recommendation of a new taper to be known as the "American Machine Taper." It is proposed that this taper be $\frac{3}{4}$ inch to the foot or 1:16. The Committee voted to prepare and send out a questionnaire to test the sentiment among the users of machine tapers. They proposed to ascertain whether it is generally believed (1) if one standard taper is favored, (2) if the $\frac{3}{4}$-inch-to-the-foot taper would meet with general approval, (3) if the $\frac{6}{10}$-inch-to-the-foot taper would be preferable, (4) if two standards are thought to be necessary, and (5) if so, which two would be preferred. This questionnaire is to be accompanied by a historical statement covering the development and use of the present tapers and technical arguments for the new taper of $\frac{3}{4}$ inch to the foot.

Business Meeting

PRESIDENT SCHWAB presided at the Business Meeting which he called to order at 1:30 on the afternoon of the first day. The attendance was very good and indicated that the membership as a whole had more than passing interest in the Society's machinery. One of the most important matters to be brought before the meeting was the new Publication Policy. This was presented and explained in some detail by R. E. Flanders, Chairman of the Committee on Publications. Some discussion followed, but this was in the nature of questions to clarify certain points rather than suggestions for improvements. The general effect seemed to be very pleasing, and when placed to a vote the Policy was unanimously approved.

Pittsburgh was announced as the place for the next Spring Meeting. Secretary Rice, as one of the steps in the procedure for adopting standards, presented by title the following:

Code for the Design of Transmission Shafting
Cast-Iron Pipe Flanges and Flanged Fittings for 125 lb. per sq. in.
Cast-Iron Pipe Flanges and Flanged Fittings for 250 lb. per sq. in.
Malleable-Iron Screwed Fittings for 150 lb. per sq. in.
Cast-Iron Screwed Fittings for 125 and 250 lb. per sq. in.

Having previously been submitted in draft form for revision and having been adopted by Council, these standards are now, after their presentation at a public meeting of the Society, ready for acceptance by the American Engineering Standards Committee as American Standard.

Entertainment Events

AS EVERY ONE well knows, the success of a meeting is largely dependent upon the success of the entertainment events. The technical events may be ever so good and well attended, but after all the first thought after returning to the workshop is some bit of amusement and the new acquaintances it brought. This in turn brings forth a train of reminiscences, each with its new acquaintance and exchange of experiences.

White Sulphur Springs being far from an industrial center, there were of course no inspection trips. The Entertainment Committee, therefore, had to be exceptionally active to provide sufficient entertainment to fill these gaps.

The initial event was the combined bridge and dance of Monday evening. The dance floor and orchestra were apparently quite to the liking of those who ventured forth, and if one may judge by the laughter and friendly arguments issuing from the vicinity of the bridge tables, the pleasure was not entirely confined to the vicinity of the orchestra.

Tuesday afternoon marked the beginning of a most interesting and hotly contested golf tournament for the men. The ladies enjoyed a motor tour through the hills surrounding the hotel, while both ladies and gentlemen managed to devote some time to horseback riding, tennis, archery, and swimming.

The banquet, which was held jointly with the American Society of Refrigerating Engineers, occupied the major portion of Tuesday evening. President Schwab acted as toastmaster in his usual good-natured and altogether pleasing manner. After an expression of his pleasure at the privilege of standing at the helm of such an organization, and welcoming cordially the A.S.R.E., he introduced the president of that society, W. H. Carrier, who addressed the assemblage briefly. In his remarks Mr. Carrier assured all that it was with great pleasure that he responded in the name of the A.S.R.E., for they had indeed enjoyed the pleasant and profitable association of the two Societies throughout all of their joint arrangements.

One of the most important features of the dinner was the presentation of the A.S.M.E. Medal to Wilfred Lewis for his very valuable work on gear teeth. Mr. Lewis was presented to the President by Conrad N. Lauer, Manager of the Society. His presentation remarks follow:

"Mr. President: It is my privilege and pleasure to present to you tonight for the bestowal of a great honor a man who typifies the highest ideals of American engineering, who by virtue of his achieve-

ments has contributed immeasurably to the present perfection of our mechanical knowledge, and who by the modesty of his demeanor and the friendliness of his heart has endeared himself to his fellow-engineers. I am especially pleased to have been called upon for this service, as your medalist is a friend of many years and a fellow-Pennsylvanian.

"This man is a world authority on gears. His formula, the universal guide in gear design, is the result of abstract analysis rather than of elaborate experimental tests. He was the first specialist to eliminate the rule-of-thumb in gear design and place it on a sound engineering basis. As chairman of the A.S.M.E. Special Research Committee on the Strength of Gear Teeth he is giving unremitting thought and care to the present-day problems of gear design and use.

"This noted engineer has contributed valuable engineering work to the public without thought of reward. His papers and discussions before the Society for a period of over forty years show the unselfish service he has rendered the profession and the public.

"The engineer receives but little recognition in the public eye. His achievements are hidden from view by the complexity of our industrial structures. It is fitting, therefore, that honor be done in this public way to one who without blare of trumpets and beat of drum marched to the solution of his problem that industry might be benefited and human well-being increased. It is impossible to evaluate the savings of natural resources, the reduction in human toil, which this man has made possible by his successful solutions of gear problems. We of his profession know and appreciate his contribution. We call the world to witness that his achievement is one of major importance in the advance of our civilization.

"Mr. President, on behalf of the members and Council of this Society I have the honor to present a great engineer as a recipient of the A.S.M.E. Medal, the highest award in the bestowal of the Society.

"Mr. President, I present to you Wilfred Lewis, mechanical engineer."

In his presentation remarks, President Schwab said:

"Mr. Lewis, as President of The American Society of Mechanical Engineers it is my duty, my pleasure, and my delight to officially hand you this medal, given you by the Society for your distinguished services. May I say, my dear Mr. Lewis, that thirty-seven years ago when you were a young engineer at William Sellers & Company and I was an engineer of the Edgar Thompson Steel Works, I little thought it would be my pleasure to stand here today as President of this Society and present you this medal. It is a pleasure I shall remember, and you have my hearty congratulations."

Mr. Lewis replied:

"Mr. President, Ladies, and Gentlemen: The honor you have conferred upon me is so overwhelming that I hardly know how to express my appreciation and thanks.

"As an experimenter and designer of machine tools for many years with William Sellers & Company, I naturally became interested in gearing and belting as important factors in the construction and operation of their products. But since then, for a somewhat longer period, I have been engaged in the manufacture and sale of products which require no gearing at all and in the operation of which belting plays no direct part.

"The continuation of my interest in these specialties of my younger days has not, therefore, been inspired by the needs of my business, nor by any prospect of pecuniary gain. It has, in fact, diverted my time and attention from other matters of more serious import to the work in hand, but it has been a labor of love, and as such, it has added to the enjoyment of life, in itself an ample reward, to which your generous recognition comes as a bonus for which I am profoundly grateful.

"It is always a pleasure to feel that worth-while results have been attained by one's patient and persistent efforts in any direction, and there can be no greater satisfaction than that which comes from recognition by professional associates and friends."

The chief speaker of the evening, Mr. Loughnan Pendred, editor of the well-known British publication, *The Engineer*, proved a most interesting addition to our list of pleasantly remembered after-dinner speakers. Mr. Pendred expressed his great delight at the manner in which he had been received throughout his tour of this country, and especially that he had lost but ten minutes from late train service during his entire tour. He had learned much, he said, and he was genuinely sorry that the time had come for him to return to the "little jewel set in the silver sea," for in going he was leaving many good friends. Before closing his remarks, Mr. Pendred presented to the Society engrossed greetings from the Institution of Mechanical Engineers, of whose Council he is a member, as follows:

"To The American Society of Mechanical Engineers, assembled at White Sulphur Springs on May 24, 1927, the Institution sends greetings and heartily desires that the cordial relationship which has heretofore existed between the two societies may ever be maintained. The Institution is confident that the union of the English-speaking engineers of all countries will promote industry and increase the happiness, contentment, and welfare of mankind."

President Schwab replied as follows:

"Mr. Pendred, on behalf of The American Society of Mechanical Engineers, we accept this testimonial from your English Society and express to you our warmest appreciation of the sentiments so kindly expressed. Will you convey to them for us that appreciation and that heartfelt sincerity with which we express it? It has been a pleasure to have had you among us, and it is but a step forward in the cementing of the happy relations between our two great countries that must bear fruit in the future."

Following the dinner a water carnival in the swimming pool and dancing in the ball room occupied the attention of the guests for the remainder of the evening.

Wednesday's entertainment was made up of golf, marking the close of the tournament, a novelty golf game, which proved a regular "side splitter," a putting contest for the ladies, archery, swimming, etc. Prizes for the winners of the various contests and tournaments were awarded during the dinner hour amid much merriment. Bridge and dancing again filled the evening.

Technical Sessions

WITH the exception of the first day, all sessions were held in the morning, the remainder of the day being devoted to sports and other forms of recreation. This plan worked out admirably, if one may judge by the numbers present at the sessions and the enthusiasm with which the discussion progressed. There was a delightful informality throughout which proved most profitable to those in attendance. This is not hard to understand after all, for it is indeed most difficult to remain in one's chair and silently accept a statement that needs amplification when one has trod the golf course or taken a plunge in the swimming pool with the speaker—the challenge must be accepted. Eleven sessions were held during the four days of the meeting, as follows: Education and Training for the Industries; Central-Station Power; Fuels; Management; Wood Industries; Hydraulics; Machine-Shop Practice; Oil and Gas Power; Railroad; Industrial Power; and Materials Handling.

Session on Education and Training for the Industries

John T. Faig,[1] Chairman of the Committee on Education and Training for the Industries, presided at the session devoted to this very important subject. Three papers were presented at this session, as follows: The General Motors Institute of Technology, by Albert Sobey;[2] A Description of the Henry Ford Trade School, by Frederick E. Searle,[3] and A Uniform Apprenticeship Certificate for Coöperating Groups of Employers, by Wm. S. Conant.[4]

The entire group of papers appeared in the Mid-May, 1927, issue of Mechanical Engineering, prior to their presentation. The paper by Sobey gave particulars of an educational program designed to provide the General Motors Corporation with a supply of highly skilled service men and trained workers for the plants, and to furnish a broad technical training for its future executives. Mr. Searle discussed and gave particulars of a successful, self-supporting trade school with 1700 students who work two weeks out of three in the school shop, devote the third week to a course of study, and who each receive from $450 to $1000 annually. Mr.

[1] President, Ohio Mechanics Institute, Cincinnati, Ohio. Mem. A.S.M.E.
[2] General Motors Institute of Technology, Flint, Mich.
[3] Superintendent, Ford Trade School, Dearborn, Mich.
[4] Washington, D. C. Mem. A.S.M.E.

Conant urged all companies and schools engaged in industrial education to coöperate toward securing greater uniformity in training for each trade, and a more explicit statement of accomplishment in the certificate award.

Prior to the presentation of the papers, Chairman Faig offered a few words on the work of the Committee, mentioning especially the group of papers recently issued by the Society under the title Education and Training for the Industries. The first two papers were presented before throwing the meeting open to discussion, otherwise the program was strictly adhered to. Mr. Searle's paper was presented by R. L. Sackett[5] who also offered written discussion of Mr. Sobey's paper. Regarding the use of the title "Industrial Engineering" for a certain course, Dean Sackett said that while the course contained the usual elementary drawing, mathematics, physics, and chemistry, these subjects constituted much less than the usual course in Industrial Engineering. The studies outlined in Fig. 2, while more specialized than those of the preceding course, also fell far short of the usual curriculum in Industrial Engineering, he felt. The courses were admirably adapted to the purpose in hand, but the titles gave an erroneous idea of the content when used outside of this particular school.

The value of the educational program described could not be over-emphasized, he said, and added that this was a pressing need in this country for others of a similar character. The public schools had confined themselves largely to general preparation, and were fairly well standardized along this line, but the changes in industry had brought such a wide variety of demands that they had been almost wholly unable to meet them. Industrial education at present, he concluded—lacked the necessary vehicles for the preparation of men in industry for their jobs and for a better job. This must come very largely through the initiative of the industries themselves and a realization of the fact that responsibility and cost must be assumed by them.

Discussing Mr. Sobey's paper orally, Dean P. F. Walker[6] wished to know how the plan affected labor turnover in the plant. A second question was, "How small may an industry be and hope to be able to put a training program of this kind into effective use?"

Luther D. Burlingame[7] spoke of the general reluctance of the boys who have progressed to the higher grades in school to consider anything but a "white-collar job."

The Chairman wished to know if it was the purpose of the school to confer degrees, and also wished to know the number of graduates per year in the Coöperative Engineering Course.

E. H. Neff[8] said that the chief trouble with our educational system was that we gave little attention to the question, "What are you going to do?"

In his closure, Mr. Sobey replied to several of the questions asked. First, he explained the use of the word "industrial" as a word attached to the courses to distinguish them from college courses. Next year, he said, they would be called "fundamentals of engineering" with an explanatory clause "this is not a college course." One of the troubles encountered was the tendency of industry to pull the students away from their work too soon, giving them the impression that they were finished. This had been corrected by an understanding that when a student was taken for work he was to continue his studies until they were completed. Replying to Dean Walker's question regarding size of industry, he said that they give 80 to 90 separate courses every term. In a community with a smaller number it should be organized on the basis of something similar to a coaching and correspondence course. Regarding the entrance requirements, he said that they started pupils from the fifth and sixth grades in night school if they had the ambition and knew there was no limit to where they could go. They preferred getting local boys, but did not object to taking every one. Regarding the matter of degrees, he said that the students came to them with the definite understanding that they were to be trained for a definite type of work. While they had made it a point to train them in the manner of colleges, and were placing them on a strictly college grade, they were not prepared to state whether or not it was a degree grade of course. They were, however, checking up on the relationship between their program and the regular college training, and the matter of awarding degrees would be decided later.

J. H. Maguire[9] presented the problem of a town of 33,000 inhabitants with no means for teaching the young the things they should know in the industrial field. Mr. Sobey suggested making the proper contact with the local high-school authorities, and working out with them a program to meet the needs of the people, or if the prospective students were too scattered to get in touch with one of the better types of correspondence schools.

In the discussion following Mr. Conant's paper it was shown that there was a definite desire on the part of many of the smaller manufacturers to group together to find the proper man to supervise the work. There was also a question as to whether or not public-school graduates should be accepted without further preparation. Mr. Conant felt that it was a problem for the supervisor of the group of manufacturers to decide. If the preliminary school training was not sufficient, it must be supplemented. Prof. Faig felt that the trouble with many schools was that instead of tying up with existing organizations and insuring permanency they lasted only as long as the enthusiasm and determination of the founder lasted. He suggested that every public-service organization have in it some one to know all the educational opportunities offered in the vicinity, and know how to assemble them in courses for the training of men for a certain purpose.

L. C. Marburg[10] greatly favored keeping specialized training within the industry itself, leaving the colleges free to handle the more general courses. He also mentioned the great turn over in labor where trade courses did not prevail. The attitude of the trained man, he pointed out, was that he was a specialist, and would not be shunted about from one job to another.

J. E. Goss,[11] in a letter, said that there was too little regard for an economic balance of training. However, he pointed out, the small-shop manager was not wholly to blame for this attitude. He needed a guide in the starting and conduct of community apprenticeship. The job must be handled by someone who could give more time to it than a shop manager.

Referring to the increasing number of high schools, etc., establishing shops and laboratories, E. H. Neff suggested that the Committee on Education and Training for the Industries do something to promote the training of men properly qualified to teach in such institutions.

C. K. Tripp[12] mentioned a plan under which the boys were made to assume a certain amount of teaching as they progressed, thereby cutting the number of instructors, but Mr. Maguire did not think this advantageous to the student.

Dean Sackett said that often the high school could give at no expense the mathematics and other subjects as far as needed, and then the industrial school could take up the specialization and continue the program, which according to Chairman Faig would insure continuity of the program when if it were kept in one firm alone might die out in interest.

Session on Central Station Power

The session on Central Station Power, held under the auspices of the Power Division, also fell on Monday afternoon. The Chairman, W. L. Abbott,[13] calling the session to order at 2:30 o'clock. Three papers were presented. The first, High-Pressure Steam at Edgar Station, by I. E. Moultrop[14] and E. W. Norris,[15] discussed the design of a generating plant of the Edison Illuminat-

[5] Dean of Engineering, Pennsylvania State College, State College, Pa. Mem. A.S.M.E.

[6] Dean, School of Engineering, University of Kansas, Lawrence, Kans. Mem. A.S.M.E.

[7] Industrial Superintendent and Patent Expert, Brown & Sharpe Manufacturing Co., Providence, R. I. Mem. A.S.M.E.

[8] New York Manager, Brown & Sharpe Manufacturing Co., New York, N. Y. Mem. A.S.M.E.

[9] Superintendent, Haynes Stellite Co., Kokomo, Ind. Mem. A.S.M.E.

[10] Secretary-Treasurer, Marburg Brothers, Inc., 90 West Street, New York, N. Y. Mem. A.S.M.E.

[11] Superintendent of Apprentices, Brown & Sharpe Manufacturing Co., Providence, R. I.

[12] Superintendent of Apprentice Department, General Electric Co., Lynn, Mass.

[13] Past-President, A.S.M.E.; Operating Engineer, Commonwealth Edison Co., Chicago, Ill.

[14] Chief Engineer, Construction Bureau, Edison Illuminating Co. of Boston, Boston, Mass. Mem. A.S.M.E.

[15] Engineer, Mechanical Division, Stone & Webster, Inc., Boston, Mass. Mem. A.S.M.E.

ing Co. of Boston which uses steam at 1200 lb. per sq. in. Problems met in developing the station, equipment of the original station and its extensions, and the results obtained were presented. The second paper was by Alex D. Bailey,[16] and dealt with problems encountered in operation of Crawford Avenue Station of the Commonwealth Edison Co., Chicago, a station designed for 550 lb. steam pressure at the turbines and 725 deg. fahr. temperature. The third paper, by Everett O. Waters[17] and J. Hall Taylor,[18] was entitled The Strength of Pipe Flanges, and gave approximate methods of determining strength, proposed formulas for strength and deflection of rings and hubbed flanges, and discussed tests on hubbed flanges. Considerable interest was manifested in the papers, and a wealth of discussion, both written and oral, followed their presentation. Much of the discussion revolved around the matter of castings, their porosity, and interesting methods of porosity detection and control were presented. Pulverized fuel also came in for a great deal of comment. Considerable interesting discussion also followed the paper on pipe flanges, there being several written contributions and much oral comment. A complete account of the discussion of the session will appear elsewhere in the Society's publications during the next few months.

FUELS SESSION

The Fuels Session, held under the auspices of the Fuels Division, was called to order at 9:30 on Tuesday morning, May 24, by J. Van Brunt,[19] chairman of the Division. But one paper was presented at this session, but the great interest in the subject under discussion, and the manner in which it was handled by the author made the session one of the most interesting of the meeting. The paper, Economics of Coal Carbonization in the United States, by Geo. A. Orrok,[20] consisted of a very interesting review of the progress of carbonization in America, the attitude of industry and the general public, coal supplies, and markets for gas and other products, and brought the subject up to date in an interesting manner. A written discussion by H. D. Savage[21] followed the paper, in which the writer commented at length on the author's assumption as to the financial and economic status of low-temperature distillation at the present time. He offered a somewhat more optimistic view of the situation than was presented in the paper. Others commenting on the economic status of the process and its probable future offered their remarks from the floor. In his closure, Mr. Orrok stated that in his opinion the problem was a commercial one; that there must be a market and a sufficient price before developments on a commercial scale could be expected. A complete account of the session will appear in an early issue of MECHANICAL ENGINEERING, when both the paper and the discussion with closure will appear in full.

MANAGEMENT SESSION

C. N. Lauer[22] presided at the Session on Management, which convened at 9:30 on Tuesday morning. The session was held under the auspices of the Management Division. A paper by L. W. Wallace[23] furnished the basis for much interesting comment. In the absence of Mr. Wallace, J. E. Hannum[24] presented the paper, setting forth data obtained in a nation-wide study conducted by the A.E.C., and analyzing accident-production statistics of plants in 20 basic industries covering 120 products. In the ensuing discussion L. D. Burlingame commented at some length in writing, following which there was a period of general discussion from the floor, the major portion of which set forth the results of actual observation, and formed valuable material for the safety manager. This very interesting and important subject will be treated in a more thorough manner in the near future when the paper and discussion appear in MECHANICAL ENGINEERING.

WOOD INDUSTRIES SESSION

The Wood Industries Session, held under the joint auspices of the Wood Industries and Materials Handling Divisions, was called to order at 9:30 on the morning of May 24 with Ralph E. Flanders,[25] in the chair. Two papers were presented at this session, both of which appeared in the Mid-May issue of MECHANICAL ENGINEERING and were as follows: Materials Handling Between Stump and Board, by Landon C. Bell,[26] and The Design of Motor-Bus Bodies, by L. C. Josephs, Jr.[27]

Mr. Bell's paper dealt with processes employed in the Appalachian Mountain hardwood section where problems differ from those encountered in lowland and swamp logging. Various points considered by bus purchasers were dealt with by Mr. Josephs. Among these were utility, appearance, comfort, strength, lightness, ease of repair, etc.

Following the presentation of Mr. Bell's paper a very interesting written discussion was submitted by John Raine[28] in which he commented on the difficult terrain over which logging operations in the Appalachians were conducted. Prior to the use of mechanical power, he said, man simply felled and burned the timber which he found impeding his progress in settling the lands, thereby destroying areas badly needed later. Now, however, power and mechanical equipment played a great part in logging operations in these same regions. The writer pointed out that the decision as to the type of equipment to be used, and the responsibility for the success or failure of the operations was in the hands of the logging manager, who must be conversant with quality and quantity and equipment available.

The decision made, the enterprise launched, he continued, there must follow the application of all forms of mechanical devices if the operation is to be successful.

The most essential factor in the harvesting of Appalachian hardwoods was the railroad, and this should be the standard-gage type. The best mechanical agent that they found for the grading of such roads he said, was the gas-electric shovel with a dipper capacity of $^3/_4$ yd., operated by one man and a helper to dress up the grade behind the shovel. The machine would grade in western Greenbrier approximately one mile per month, he said.

Regarding skidding, he pointed out that the operator could not confine himself to any single method, for different hills, valleys and streams often required different means for economical operation.

He did not recommend ballhoofing, as damage to logs in their downward flight on the mountain might result.

Overhead-cableway transfer in some cases had been found very useful and effective, but its use was not recommended where a combination of rail and other mechanical means could be used. For slight uphill hauls and flat terraces the tractor was often the best means available for the skidding of logs to the rail. It should be used in conjunction with horses that made up the trails where the logs lay, the tractor acting as a transfer machine between such made up trails and the landings.

There were situations in the Southern Appalachians, he pointed out, where incline logging had been done successfully and where other methods could not be applied. In one installation mentioned it was possible to avoid the construction of some two miles of road as well as to avoid a $3^1/_2$-mile haul up a switch-back road and so over the mountain.

On steep ground and near rail situations, and where mechanical equipment could be applied, horse skidding was the most generally used method.

Eight-wheel, single-log-length, skeleton-frame trucks equipped with patent trip stakes for quick and safe unloading at pond and with carrying capacity of from 3000 to 3500 ft. of logs, seemed the most practical and economic means of transportation.

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[16] Superintendent, Generation Stations, Commonwealth Edison Co., Chicago, Ill. Mem. A.S.M.E.

[17] Assistant Professor of Mechanical Engineering, Yale University, New Haven, Conn. Assoc-Mem. A.S.M.E.

[18] President, American Spiral Pipe Works, Chicago, Ill. Mem. A.S.M.E.

[19] Vice-President in charge of Engineering, Combustion Engineering Corporation, New York, N. Y. Mem. A.S.M.E.

[20] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[21] Vice-President, Combustion Engineering Corporation, New York, N. Y. Mem. A.S.M.E.

[22] General Manager and Treasurer, Day & Zimmerman, Inc., Philadelphia, Pa. Mem. A.S.M.E.

[23] Executive Secretary, American Engineering Council, Washington, D. C. Mem. A.S.M.E.

[24] Director of American Engineering Council Investigation on Accident and Production, Washington, D. C. Mem. A.S.M.E.

[25] Manager, Jones & Lamson Machine Co., Springfield, Vt. Mem. A.S.M.E.

[26] W. M. Ritter Lumber Co., Columbus, Ohio.

[27] Engineer, International Motor Co., Allentown, Pa. Mem. A.S.M.E.

[28] President and General Manager, The Meadow River Lumber Co., Rainelle, W. Va.

For his company's mountain work a 70-ton geared locomotive had proved most practical and effective.

They did not use nor approve the stiff-boom type of loaders illustrated in the paper, but used instead the more mobile swinging-boom type with a line radius of about 100 ft. The patented trip stake doing away with the use of chains had been found the safest and most economical method of transporting and quickly unloading logs in the pond.

The writer favored electric stackers at the mills, and even electrically propelled trucks for dry-kiln and planing-mill operation.

Further comment was offered in writing by B. B. Burns,[29] who said that in the past his company has used larger types of skidders on a more extensive scale than contemplated by Mr. Bell, but had found it unprofitable for many reasons, the principal one being the scarcity of timber on the ground, the available land being timbered to 5000 ft. per acre, and for the further reason that heavy machines required a more expensive type of railroad than generally used.

F. W. Shawhan,[30] also in writing, pointed out that mechanical equipment and methods used should possess sufficient reserve capacity to meet the stresses imposed in moving of logs up to 60 in. in diameter and weighing up to 10,000 lb.

The forces employed were enumerated as follows: (*a*) gravity; (*b*) animal power; (*c*) gas; (*d*) steam.

In team skidding, he said, gradient was the controlling factor, and as uniform grade as possible was desirable.

As to the relative merits of the three motive forces, animal, gas, and steam, as expressed by horses, tractors, and steam skidders, respectively, each had its advantages and each had its marked limitations, therefore, each must be carefully studied. For instance, teams were mobile and could haul loads around corners and curves; tractors were effective in hauls out of a hollow; steam skidders were limited to an effective hauling distance of 3000 ft., could not haul logs around corners at a reasonable cost, but must work in a straight line, but usually beat teams, for unit cost, within their working radius, in a stand of timber of 6000 ft. per acre, or better.

The choice of narrow-gage or standard-gage tramroad, the writer said, usually depended upon such considerations as topography, haulage or by-products, etc.

Geared engines were used almost exclusively, negotiating grades up to a normal maximum of eight per cent, and, in special cases, up to 12 per cent.

At this point the meeting was thrown open to oral discussion. In reply to a question by Mr. Josephs, regarding permanence of operations, Mr. Bell stated that about 60 per cent of the value of the timber in the Appalachians was produced from about 30 per cent of the trees that were over 200 year of age. Therefore, it was quite impossible to continue after installing a hardwood plant until lumber grew again. In some of the soft woods, however, it was possible to re-grow salable timber within 20 to 30 years, in which case reforestation might be carried on profitably and the plant maintained.

There was also some discussion of the subject of taxation, the conclusions being that it was unfair to tax property bearing trees year after year, the preferable plan being to tax on a yield basis when the timber was cut.

Following the paper by Mr. Josephs, the matter of bracing wood motor-bus bodies was discussed. Mr. Josephs' contention was that metal parts were necessary for bracing; gluing and mortising being insufficient due to the twisting action imposed upon the bodies.

Mr. Bell discussed at length the use of wood in the face of popular belief that it was more dangerous from the fire standpoint than metal. He related several experiences showing that in many cases, wood although inflammable, was conducive to less destruction than metal in the case of fires. He felt that a mixture of the two commodities was necessary, and favored wood in automobile or railway car for comfort.

F. P. Cartwright[31] said that contrary to popular belief, our lumber supply was not dangerously near to depletion—merely certain types of woods were getting low in supply. He also mentioned the fact that lumber was a crop and not a mine. Reforestation, he said, must be predicated on the probability that there would be a large market for that lumber when grown and that it would justify prices proportional to the cost of production.

The Chairman spoke at some length on the general matter of the economics of deforestation and reforestation. He also metioned the matter of price. Discussing the situation in Germany and other European countries he said that scaffolding, for instance, was constructed of peeled poles carefully erected and then removed after each job and preserved for further use. He wished to know why the hardwood industry in New England had not been developed to a greater extent. Mr. Shawhan replied that the New Englander had always looked with more or less contempt on hardwood. It was hard for him to get it sufficiently consolidated for the installation of a plant, railroad connections were bad, and the wood would not float to the mill.

Mr. Bell mentioned the impression on the part of the public that the lumberman was a ruthless waster of the forest, and stated that, using the word in the proper sense, he knew of none of the Appalachian forests which had been wasted. A lively market for by-products would aid greatly in saving and utilizing small logs and the usual waste of branches, etc. The average person, however, not realizing the lack of demand for these by-products believed that all of this represented intentional waste.

Mr. Bell also mentioned the subject of standardization and recommended that proper steps be taken to standardize lumber sizes as a means of preventing waste.

Mr. Cartwright added that the greatest difficulty lay with the consumer, who would not realize the possibilities of grading rules. He favored a rather extensive program of research in the properties of lumber, and acquainting the consumer with the findings.

HYDRAULIC SESSION

R. L. Daugherty[32] presided at the Hydraulic Session, held under the auspices of the Hydraulic Division of the Society, which convened at 9:30 a.m., May 25. Three papers were presented at this session, all of which appeared in the Mid-May, 1927, issue of MECHANICAL ENGINEERING. The first, by W. S. Lee,[33] on Rack Structures and Headgates of Cedar Creek Hydroelectric Plant, discussed details of equipment adopted for the water intake at this station of the Duke Power Co. on the Catawba River near Great Falls, S. C. Arnold Pfau[34] presented the second paper of the morning under the title, Specific Characteristics for Hydraulic Turbines, suggesting a new expression for specific speed that was non-dimensional and embraced the three fundamental characteristics of a turbine, namely, peripheral speed coefficient, velocity of flow coefficient, and efficiency. The final paper on the program was Comparison and Limitations of Various Water Hammer Theories, by Ray S. Quick.[35] Mr. Quick offered confirmation experimental test data on the accuracy of the elastic-water-column theory. He gave charts for graphically solving the problems of maximum pressure rise with uniform gate motion and complete closure, and other information of value to the hydraulic engineer. Also scheduled for a place on the program was the Progress Report of the Division to be presented by Lewis F. Moody.[36] This, however, did not receive any discussion, owing to the lack of time available.

Considerable discussion was offered on both the Pfau and Quick papers, much of it being in written form and involving calculations and diagrams. The oral discussion proved most interesting and brought to light considerable information of value in the hydraulic engineer's sphere. Owing to the length of the discussion, it is impossible to do it full justice in abstract; therefore, it will be treated fully elsewhere in the Society's publications at a later date.

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[29] Secretary-Treasurer, C. L. Ritter Lumber Co., Huntington, Del.

[30] Beckley, W. Va.

[31] Technical Representative, National Lumber Manufacturers' Association, Washington, D. C. Assoc-Mem. A.S.M.E.

[32] Professor Mechanical Engineering and Hydraulic Engineering, California Institute of Technology, Pasadena, California. Mem. A.S.M.E.

[33] Vice-President and Chief Engineer, Southern Power Co., Charlotte, N. C. Mem. A.S.M.E.

[34] Consulting Engineer, Hydraulic Department, Allis-Chalmers Manufacturing Co., Milwaukee, Wis. Mem. A.S.M.E.

[35] Engineering Department, The Pelton Water Wheel Co., San Francisco, Calif. Assoc-Mem. A.S.M.E.

[36] Consulting Engineer, I. P. Morris Corporation, Philadelphia, Pa. Mem. A.S.M.E.

MACHINE-SHOP-PRACTICE SESSION

Papers on mechanical springs, arc welding, and electrical machines were presented at the Machine Shop Practice Session, held under the auspices of the Machine Shop Practice Division, which convened at 9:30 a.m., May 25. F. H. Brown[37] presided.

The first paper was presented by Joseph K. Wood[38] and dealt with Hysteresis Relative to the Operation of Mechanical Springs. Mr. Wood showed that springs, and spring systems in particular, were associated with physical hysteresis of three characteristic types, namely, mechanical, hypo-elastic, and hyper-elastic. Elastic hysteresis effects are sufficiently large to be given consideration in the design of measuring devices.

J. F. Lincoln[39] presented the second paper under the title Arc Welding, setting for advantages of arc-welded steel parts over those of cast iron or riveted construction, and showing the savings in cost effected by replacing iron castings with arc-welded steel parts, the inconsistency of permitting arc welding in superheaters and steam piping and forbidding it in boilers, etc.

With the present tendency toward the elimination of noise, the third paper, by J. P. Den Hartog,[40] on Vibration of Frames of Electrical Machines, proved most interesting. Mr. Den Hartog gave formulas and curves for calculating natural frequency of vibration, and showed that frames which usually emit a large portion of the total noise can in many cases be regarded as a part of a ring with rigid ends.

Considerable discussion followed the presentation of the papers in which both written and oral comments were included, but the amount of material is too great to be abstracted here, and it will be necessary to reserve space in a later issue of MECHANICAL ENGINEERING. The paper by Mr. Den Hartog will also be treated more fully elsewhere at a later date.

OIL AND GAS POWER SESSION

L. H. Morrison,[41] Secretary of the Executive Committee of the Oil and Gas Power Division, presided at the session under that Division's auspices on May 25. The session was called to order at 9:30 a.m.

The first paper presented was that on The Study of Oil Spray for Fuel-Injection Engines by Means of High-Speed Motion Pictures, by Edward G. Beardsley,[42] which described the apparatus developed on the laboratory at Langley Field for photographing the start, growth, and cut-off of oil sprays from injection valves. This investigation was carried on particularly to determine the adaptability of the oil-injection type of engine to aircraft.

Because of the similarity of the two papers, that by Carlton Kemper[43] on Experimental Combustion Chambers Designed for High-Speed Diesel Engines was presented before throwing the meeting open to discussion. Mr. Kemper's paper discussed cycles for high-speed fuel-injection engines, three types of combustion chambers, and results of tests showing performance of engines equipped with each of the three types. The lengthy discussion following these two papers indicated unmistakably the keen interest being aroused in the solid injection type of engine for high-speed work. Both of these papers will have fuller treatment elsewhere in the Society's publications, along with both the written and oral discussions.

Some Uses of the High-Speed Multi-Cylinder Indicator was the title of the paper by H. M. Jacklin.[44] Mr. Jacklin gave particulars of a simple, rugged device for use on high-speed internal-combustion engines and air compressors; multi-unit development for obtaining diagrams simultaneously from all cylinders of engine or compressor. This paper called forth considerable written discussion, of too great length to be abstracted in this space. Therefore the paper and discussion will receive treatment at length in an early issue of MECHANICAL ENGINEERING.

A paper on The Range and Severity of Torsional Vibration in Diesel Engines, by Frederic P. Porter,[45] was read by title. This paper will also appear later in MECHANICAL ENGINEERING.

RAILROAD SESSION

Two papers dealing with two types of locomotives were presented at the Railroad Session. William Elmer,[46] Member of the Executive Committee of the Railroad Division, under the auspices of which the session was held, presided. The meeting was called to order at 9:30 on the morning of May 26.

The first paper dealt with High Steam Pressures in Locomotive Cylinders. In this paper, the author, Lawford H. Fry,[47] surveyed efficiencies obtainable with various steam pressures, and examined the effects of the ratio of expansion on efficiency. His conclusions were that it was impossible to secure a considerable increase in thermal efficiency of cylinders by increasing boiler pressure. William Arthur[48] presented the second paper, treating the subject Diesel Traction for Railroads. A general discussion, covering advantages of "Dieselizing" certain railroad services; operating, design, and construction problems; weights and speeds; cooling, transmission, and vibration problems; maintenance; fuel consumption, etc., was given. He favored the adoption of Diesel-electric traction.

Both papers appeared in the Mid-May, 1927, issue of MECHANICAL ENGINEERING. Considerable discussion followed each paper, and great interest in the two very different subjects was indicated. The first written discussion was by R. Eksergian,[49] who said that the locomotive cycle used by Mr. Fry gave a close approximation to that of an actual cycle. Actual performances, however, required the use of a cycle factor, which must take care of (1) the equivalent ratio of expansion with a given clearance volume, (2) the cooling (condensation effect) of the steam during admission in the clearance space, (3) the wire drawing and throttling and the corresponding decrease of availability, (4) the reheating factor which modifies the expansion curve from an adiabatic, and (5) the proper compression and its effect on the card, etc.

Regarding the author's statements concerning the advantages of expansion in two cylinders rather than one, he said that it was interesting to compare such expansions. With simple cylinders, with increasing expansion ratios, the cut off necessarily must be reduced until we reached the condition where the percentage of clearance volume to cut-off volume became so large that the loss due to cooling during admission would offset the gain due to expansion, definitely limiting as to given expansion ratios with simple cylinders. The expansion ratio being at best relatively small for any ideal cycle, the clearance volume itself, therefore, had a marked effect on the ratio of expansion, he said. A further disadvantage of single cylinder expansion was the abnormal ratio of peak to mean piston load, if advantage was to be taken of better expansion ratios. This, therefore, required heavier machinery at the expense of a smaller boiler with poorer counterbalance conditions.

Compounding offered advantages in the use of greater expansion ratios, which were needed to take care of the higher pressures. He said an outstanding advantage was the greater range of economical performance, because (1) a reasonably good expansion ratio could be obtained at low speeds, (2) the possibility of relatively greater expansion ratios due to the fact that the cooling effect of the clearance space was reduced. The throttling loss was increased over that of a simple cylinder, but the greater gain in better expansion ratio overbalanced this.

H. B. Oatley,[50] also in writing, mentioned the fact that the question of pressure must go along with the question of tempera-

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[38] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[39] Vice-President, Lincoln Electric Co., Cleveland, Ohio.

[40] Research Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

[41] Associate-Editor, *Power*, McGraw-Hill Co., Inc., New York, N. Y. Assoc-Mem. A.S.M.E.

[42] Junior Mechanical Engineer, National Advisory Committee for Aeronautics, Langley Field, Va.

[43] Junior Mechanical Engineer, National Advisory Committee for Aeronautics, Langley Field, Va.

[44] Assistant Professor of Automotive Engineering, Ohio State University, Columbus, Ohio. Mem. A.S.M.E.

[45] Technical Aide, Brooklyn Navy Yard, Brooklyn, N. Y.

[46] Special Engineer, P.R.R. Co., Philadelphia, Pa. Mem. A.S.M.E.

[47] Metallurgical Engineer, Standard Steel Works Co., Burnham, Pa. Mem. A.S.M.E.

[48] Traction Division, American Brown Boveri Electric Corporation, Camden, N. J.

[49] Engineer, Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

[50] Vice-President in charge of Engineering, Superheater Co., New York, N. Y. Mem. A.S.M.E.

ture. He admitted, however, that the practical problem of reheating was far harder for the locomotive designer with the limited space available, than for the industrial-plant designer.

He mentioned briefly the Schmidt-Henschel double-pressure locomotive built to the order of the German State Railways, a locomotive of more than ordinary interest because of the marked advance toward higher steam pressures; carrying, as it did, a working pressure of between 850 and 900 lb. per sq. in., and also because of the double-pressure arrangement of the boiler. He gave also a brief presentation of some of the data showing the performance of this locomotive, which supported the thermal analysis which Mr. Fry had given.

The locomotive was, in comparison with two American engines illustrated, a very small unit, and it might reasonably be expected that had it been of a size comparable with the modern American locomotive, there would have been obtained still better results.

As was to have been expected, a number of difficulties were encountered in this first double-pressure locomotive, but all were of a relatively minor character. Boiler-feeding devices originally applied were not entirely suitable, and the means for indicating the water level required some modification. The performance of the locomotive showed these details had been made suitable and that this system of steam generation and use was a perfectly satisfactory, safe, and efficient design. No changes were necessary in the method of handling the locomotive, and the engine crews had no difficulty in adapting themselves to its operation.

The results of the tests fell short of the designed best performance, however, and some changes were being made which were expected to provide an additional 10 per cent in fuel economy.

The indirect method of steam generation utilized in this locomotive had been used for some time in power plants for stationary service, and offered advantages described in part as follows:

1 The high-pressure drum was not in direct contact with the high-temperature gases

2 The steam generation in the high-pressure drum was uniformly distributed throughout the entire length of the drum, eliminating so-called "geyser action," and resulting in a smaller moisture content in the steam.

3 The circulation of water through the indirect heating system was influenced only by the temperature conditions in the firebox

4 No stayed surfaces were used either in the indirect heating system or in the high-pressure steam drum

5 Firing up of a boiler with the indirect heating system could be done more rapidly and with little stressing of the boiler material

6 The heating surfaces in the high-pressure drum were practically scale-free, and cleaning of the high-pressure boiler consisted simply in washing out any soft sludge which might accumulate.

During the oral discussion of Mr. Fry's paper Geo. A. Orrok stated that the trouble with the high-pressure locomotive did not lie in the engine parts but in the boiler. The track gage definitely determined the size of the low-pressure cylinder, which in turn determined the pressure possible to carry in the locomotive, with economy.

The boiler, on the other hand, involved the matter of type of construction. Locomotives running anywhere from 40 to 60 m.p.h. over a rigid track with humps and hollows and rail joints received shakes not conducive to high-pressure boilers. Referring to the work of Messrs. Schmidt and Hartman, as exemplified in the Schmidt-Henschel locomotive, he said there was a second type of boiler known as the Löffler type, using practically the same idea. A third type, patented by Brown-Boveri eliminated the circulating pump of the Löffler type. He was pleased to note the 2 lb. of coal per drawbar hp. mentioned, which compared with the old practice of 8 or 9 lb., seemed to indicate very near the limit which might be expected.

Referring to turbine locomotives which he had observed, he expressed the opinion that they were not satisfactory commercially. The Diesel engines he had observed did not seem to offer possibilities on our 4-ft. 8 in. gage, except in very light traffic.

E. P. Ripley,[51] mentioning compound locomotives, stated that his company at one time had in the neighborhood of 800 of this type which showed economical results in operation, yet because of difficulties in operation and maintenance these locomotives were being converted to single-stage as rapidly as possible and at great expense. For Western use in bad-water regions he did not consider the present type of water-tube boiler satisfactory, owing to the length of time required for washing out.

Mr. Orrok emphasized the importance of proper cut-off to obtain maximum expansion of the steam.

In his closure Mr. Fry stated that in the type of 4-cylinder compound locomotive operated with two cylinders inside the frame and two outside, the greatest difficulty in maintenance arose from attempting to carry too great a load on a 2-throw crank axle. The present tendency toward 3-cylinder locomotives with a single-throw crank axle seemed satisfactory.

Commenting in writing on Mr. Arthur's paper, O. D. Treiber[52] stated that since the weight of the frame, driving motors, cab, and controls could not be much further reduced, the Diesel-engine designer must increase the revolutions of the generator, either through increased speed of the engine or through step-up gears, in order to reduce its weight. By the use of properly selected metals he felt it was possible to reduce the weight of a Diesel engine from 10 to 25 lb. per b.hp., depending on size, type, and arrangement. Regarding types of engines, he favored the 8-cylinder opposed type, but doubted that it could be adopted satisfactorily because of its bulk. The 8-cylinder-in-line type, because of its great length would add fore-and-aft bending moments that would require especially stiff framing. Six cylinders would introduce unbalanced fore-and-aft forces. The 4-cylinder type then appeared most feasible, although it would not be free from vibrations. He then touched upon recent developments in aluminum alloys, mentioning a metal which his company used and which cost roughly 50 cents per lb., could safely be worked at a stress of 9000 lb. per sq. in., and weighed one-third of cast iron. Steels were mentioned, which, when properly heat-treated, could give tensile strengths well over 80,000 lb. and safe working stresses around 15,000 lb. It was possible, he said, to build a Diesel engine weighing about 3 lb. per hp., working the metals under great stresses, or 20 to 25 lb. per hp. with the stresses well within reasonable limits. He briefly mentioned the cost of building light-weight engines, touching particularly upon difficult tolerances, power variations with variations in weight, etc.

Regarding expense incident to the installation of an electrification system, A. H. Candee[53] said in writing that the Diesel locomotive undoubtedly had a very decided advantage over the electrification of medium- or light-traffic lines where initial capital was difficult to obtain or where extreme power concentration was not required, but the economies of electrification had been proved in a large number of cases, whereas the Diesel was somewhat unknown as yet. Comparing the Diesel unit with the electric locomotive he said that with an equal capacity in electrical equipment it was entirely possible to secure much higher speeds with the higher accelerating tractive efforts with an electric locomotive than with a Diesel locomotive.

Regarding heating systems for Diesel locomotives drawing passenger trains, he said that plants had been built in trailers for attaching to such locomotives, but this always involved the question of additional train-men, and it had been found more desirable from the railroad's point of view to include such units in the locomotive itself.

The pioneering spirit which the railroads were showing by the purchase of Diesel locomotives was a very healthy condition, he felt, and we might expect that under this stimulus the development of the Diesel locomotive to a high degree of perfection would require a much shorter period of time than had been necessary for either the steam or the electric locomotive.

Further written comments were submitted by D. L. Bacon,[54] who declared that the next year would see in operation several Diesel

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[53] Railway Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

[54] Supervisor Auto Equipment, New York, New Haven and Hartford Ry. Co., New Haven, Conn. Jun. Mem. A.S.M.E.

locomotives far more ambitious than anything heretofore constructed in this country. Although the Diesel engine had the reputation of being inherently heavy, he felt that it was more than probable that the weights of such engines would, if there was sufficient demand, be reduced to an entirely satisfactory figure. The design problems appeared to him, on the whole, to be more easily solved than the economic ones. He mentioned an item in this connection which, in certain applications, might become the deciding factor; namely, the maintenance of isolated enginehouses with fuel, water and ash-handling facilities for one or two steam locomotives as compared with the strategic location of a few Diesel locomotives which required only cleaning and inspection at infrequent intervals. Proper use of the latter might permit of discontinuing obsolete and inefficient maintenance points.

During the oral discussion Geo. A. Orrok, referring to the matter of high speeds in Diesel engines, stated that he had run a $3\frac{1}{4} \times 5$-in. Diesel engine at 2600 r.p.m., the equivalent of about 1400 ft. per min. piston speed with an engine life of about 200 hr. He had also run steam engines at piston speeds up to 1500 ft., but their lives were very short. Diesels at 1600 to 1800 ft., and in some cases 2000 ft., piston speed had extremely short lives. For good life in a railway business he recommended not exceeding 1100 ft., and better 900 ft. A limit of 750 ft. had been found best for steam service.

E. P. Ripley saw the great advantage of the Diesel engine over the steam locomotive in regions where water was extremely difficult to obtain and of poor quality.

In his closure Mr. Arthur stated that although the Diesel locomotive had kept the electrical designer guessing, he felt that with the improvements in the steam locomotive it would be a long time before the latter disappeared from our lines. He thought, however, that within five years Diesels would be built up to 3000 hp. in one unit, when they would be quite comparable with some of the better forms of steam and electric traction.

Industrial Power Session

The Industrial Power Session, held under the auspices of the Power Division of the Society, was presided over by John H. Lawrence.[55] The session convened at 9:30 on the morning of May 26.

The first paper to be presented was by Henry F. Scott,[56] on Management of Industrial Power—The Plant Engineer's Viewpoint. Mr. Scott told in a simple manner of the relations between the plant engineer and the management, avoiding technical details and describing the manner in which successful results can be obtained if there is common understanding of the problems of the plant by the plant engineer and the management. Written comment was offered at this point, but the oral discussion was postponed until after the presentation of the paper on Management of Industrial Power—The Manager's Viewpoint, by Walter N. Polakov.[57] This paper pointed out that the executive should know the possibilities of his installations and if they were being fulfilled. He should also know the advantages of outside power, the cost of power interruption and the apportionment of power costs to different products.

Both papers were now thrown open to discussion, which proved most interesting and added greatly to their value. A complete report will appear in an early issue of MECHANICAL ENGINEERING.

The final paper on the program was presented by W. L. Badger[58] and discussed the subject of Evaporators for Boiler Feed Make-Up Water. A general discussion of evaporator types and evaporator operation as they would be involved in the production of relatively large amounts of boiler-feed make-up was presented. Factors affecting rates of heat transfer, the general problem of foam and entrainment, and basic principles of multiple-effect operation were included. As in the case of the first two papers of the session, Mr. Badger's contribution will receive further treatment in MECHANICAL ENGINEERING.

[55] Vice-President and Engineering Manager, Thomas E. Murray, Inc., New York, N. Y. Mem. A.S.M.E.

[56] Plant Engineer, Dennison Manufacturing Co., Framingham, Mass. Mem. A.S.M.E.

[57] President, Walter N. Polakov & Co., Inc., New York, N. Y. Mem. A.S.M.E.

[58] Director, Evaporator Experiment Station, Ann Arbor, Mich.

Materials Handling Session

A session on Materials Handling was held under the auspices of the Materials Handling Division of the Society on Thursday morning, May 26. James A. Shepard,[59] Member of the Executive Committee of the Division, called the session to order at 9:30 a.m.

But one paper was presented at this session, namely, The Relation of Building Design to the Manufacturing Process, by Charles P. Wood,[60] which appeared in the Mid-May, 1927, issue of MECHANICAL ENGINEERING. Broad principles governing building projects were discussed, also special features of building design. Description of buildings for the manufacture of soap, linoleum, plate glass, and textiles, and for the publishing of magazines and newspapers were given. The paper proved of considerable interest, and both written and oral discussions were presented, adding considerably to the information presented. The first written discussion was presented by C. G. Spencer,[61] who laid great stress on the importance of good foundations, "the most important part of the building."

For some locations and conditions, he said, the foundation costs approached those for the superstructure. In other favored locations, the cost might be reduced well below the average. Each site must be studied for bearing value and superimposed loads. With a reasonable doubt as to the soil structure and where it was important that there be no settlement in the structure, generally the case in additions to existing plants where floor levels and conveyor and pipe lines must be maintained at elevations established in the existing structure, be considered it is essential that the underlying conditions be explored by core drilling to establish the exact depth of rock or load-bearing material, the nature of the soil through which the drill passed and the character of the bearing material when reached. As an indication of what might be expected to follow negligence in this respect, the writer mentioned results of a silo installation for a cement mill. The usual preparations of driving piles to gravel were made and the silo built. When it was loaded the settlement was in feet rather than inches. Drilling would have located the underlying strata of quicksand and mud. Also, he said, needlessly expensive foundations had been built on ground where the underlying structure was good, but this fact had not been determined by drilling.

The decisions for floor loadings affected the structural design and, as a result, the cost of the building from the top floor to the foundations. There was usually three alternatives: designing the floors for loadings sufficient to accommodate the process equipment and stored raw or finished material at any point on a floor, being the first. The next, which represented some initial saving, was to design the floor for an average loading and reinforce those areas subject to concentrated loads; but this design did not lend itself well to change in process or routing of material. The third alternative was to study the equipment and process loadings in further detail and design for the special loadings; stenciling the columns and walls of all bays to indicate the safe loading and to depend on careful supervision not to exceed this. Decisions for floor loadings in addition to manufacturing convenience and relative structural costs, must take into consideration the adaptability of the building for possible other uses or industries.

Many plants were handicapped because low ceilings in existing buildings did not permit of the installation of modern machinery. This was due to the error of overlooking the fact that the cubic foot cost for additional head room was an increment of and not the same as the cubic foot cost for the entire building.

A written discussion from the Chairman pointed out that freak features in building design rarely constituted a perfect solution of any manufacturing process, hence, if the process selected apparently called for such features, it became more than likely that the proposed process was faulty and should be more thoroughly studied and carefully revised.

He gave two general rules which should always be observed in planning industrial buildings:

1 Locate the several processes so as to promote, as far as pos-

[59] Vice-President, Engineering, Economic Research, Shepard Electric Crane & Hoist Co., Montour Falls, N. Y. Mem. A.S.M.E.

[60] Lockwood, Greene & Co., Inc., New York, N. Y.

[61] Engineering & Construction, Baker & Spencer, Inc., New York, N. Y. Mem. A.S.M.E.

sible, a natural flow of materials from process to process with the minimum of intermediate handlings.

2 Choose building details which would facilitate all established systems of materials handling which might be made applicable to manufacturing of the general character contemplated.

There were surprisingly few special features required in the observance of both of these rules, he said. He then gave the four general classes of equipment for handling materials in industrial buildings, as follows:

1 Wheeled vehicles, which might only affect building construction as to the character of floors, location and size of doors, and the disposal of aisles and passageways

2 Elevators, involving almost no special building details but intelligent location

3 Conveyors, usually applicable in all buildings of standard construction

4 Traveling overhead hoists, usually requiring special building details. The cost of these special features was relatively small, however, and since they afforded opportunity to utilize a type of handling equipment widely employed in industrial operations, he considered them desirable.

An industrial building constructed in a manner which would preclude or impair the successful employment of wheeled vehicles, elevators, conveyors, traveling overhead hoists, or any of them, represented an undesirable and uneconomical type of construction, he maintained. A building without the conveniences which had become standard for industrial buildings represented specialization in its most extreme form.

Traveling cranes or traveling monorail hoists must be placed below all ceiling obstructions within the area served, in order to conserve head room:

All floor girders over areas which might require crane or hoist service should be placed parallel to the line of travel and not crosswise. He also urged avoiding knee braces or corbels. Pipe lines should not be placed indiscriminately along or across ceiling areas. Avoid round columns whenever possible, he said, as they decrease available hoisting area. In buildings provided with a central crane bay, provision for the installation of spur tracks over the side bays should be made by making the side bays of approximately the same height as the main bay, thus permitting an increase in available area of 100 to 200 per cent if required.

H. V. Coes[62] wrote that owing to the intensive mechanization and obsolescence of equipment, it was important to have the layouts as flexible as possible with as few permanent items, such as special foundations, pits, different floor levels, etc., as possible without impairing the efficiency of the plant. The large scale upon which modern manufacturing processes were carried on, he pointed out, demanded that the best and latest type of machinery be used, therefore provision for its installation as demanded should be made, and the building for housing it must be capable of receiving that equipment.

In the initial economic studies as to location, it was important to know whether it was of greater advantage to move to an entirely new location or to expand in the old location, or to build a branch plant at a new location, he pointed out.

Hard-surfaced roads, the automobile, the truck and trailer, excellent railroad service, and other modern developments were bringing about a decentralization of industry. Concentration in large cities and congestion of freight terminals also were factors in bringing about this decentralization. The writer showed by means of a table and map that this migration of industries had reached an appreciable figure.

Regarding Mr. Wood's statement concerning the necessity of making the building and the equipment an operating unit, Mr. Coes said that many times the coördination and correlation of the building and its service equipment with the productive unit was sacrificed because the proper construction factors had not been adapted to the requirements of proper operation, or again the industrial engineer has been handicapped because of limitations of the property.

Discussing the paper orally, Irving Fellner[63] mentioned the value of air conditioning in some types of factories. He also told of an experience in a newspaper plant which illustrated the value of correct layout. A strike failed to tie up production because of the fact that extremely skilled help was not necessary; mechanical equipment filled the breach. Regarding the migration of industries, Mr. Fellner cited a case where great amounts of explosives were required at a remote location, making it necessary to ship in the supplies from two distant points. The manufacturer of the explosives saw the advantages of being on the job and built a plant in the district, which, because of the coöperation of two industries using the products, flourished. Another opportunity for a similar arrangement was in the mid-continent oil fields, he said, where great quantities of sulphuric acid were shipped in. Large quantities of chemicals were required in North and South Carolina and Georgia, yet no local plant was prepared to fill the demand. Much of the decentralization, he said, was due to the fact that the larger companies had not studied the situation and determined the advantages of branch plants.

C. H. Matson,[64] to illustrate what might happen when a company thought it knew its business and attempted to make extensions on the basis of that opinion, mentioned a plant extension for the manufacture of very special apparatus, and which, when 90 per cent complete was found to embrace an almost obsolete method. New methods and machinery had been developed and those responsible for the plant extensions were not aware of them. The new machinery would produce six times the output of the older types for which the building was designed, but could not be fitted into the space provided, consequently the building lay idle for years and was the source of considerable annoyance and useless expense to the company.

James Towart[65] lamented the tendency of many architects to envelop a proposition in architectural effects and to slight such very important details as sprinkler systems, rails for cranes and other handling equipment, elevators, piping, etc. The result usually was that the steel fabricator encountered all sorts of difficulties. Very often, he said, it was necessary to work within very narrow limits, within an inch or two, making it virtually impossible to develop the full intent of the plan and specifications without going to the architect for consultation to ascertain the ultimate need of the equipment. He further recommended that nothing be allowed to go into the building until the material handling equipment or method had been determined. E. G. Ackart[66] also emphasized the fact that architects often failed to visualize the flow of materials properly, the result being a poor plant layout and uneconomical results.

Citing the automobile industry as an example, Mr. Ackart pointed out that changes of methods or models need not worry one greatly if the factory building were so designed that machinery might be moved about easily and fitted into the new schemes as they were developed. The dye industry has also offered an excellent example of the great confusion that might result if proper attention were not given to building layout. Sudden changes in fashions demand sudden changes in manufacturing programs, and hence there was a necessity for buildings permitting unimpaired expansion or shifts of equipment.

An interesting observation of the author of the paper was that the best field for engineers was to keep the financiers from making mistakes, once they had decided to start operations. Regarding Mr. Coes' remarks on the migration of industries, he said that this was possible now, while in earlier days it could not be done because of climatic conditions, but through the use of air-conditioning apparatus it had become possible to manufacture wherever it might be desirable many of the articles formerly localized because of such conditions.

Before closing the session, the Chairman expressed the opinion that it might be possible for a committee of the Division to develop a standard type of structure for some of the more conventional types of manufacturing plants where no marked deviations would be expected.

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[62] Vice-President and General Manager, Belden Manufacturing Co., Chicago, Ill. Mem. A.S.M.E.

[63] McGraw-Hill Publishing Co., New York, N. Y.

[64] Assistant to General Superintendent, General Electric Co., Fort Wayne, Ind. Assoc-Mem. A.S.M.E.

[65] Shepard Electric Crane & Hoist Co., Montour Falls, N. Y.

[66] Assistant Chief Engineer, E. I. du Pont de Nemours & Co., Wilmington, Del. Mem. A.S.M.E.

The Conference Table

THIS Department is intended to afford individual members of the Society an opportunity to exchange experience and information with other members. It is to be understood, however, that questions which should properly be referred to a consulting engineer will not be handled in this department.

Inquiries will be welcomed at Society headquarters, where they will be referred to representatives of the various Professional Divisions of the Society for consideration. Replies are solicited from all members having experience with the questions indicated. Replies should be as brief as possible. Among those who have consented to assist in this work are:

ARCHIBALD BLACK,
Aeronautic Division
H. W. BROOKS,
Fuels Division
R. L. DAUGHERTY
Hydraulic Division
JAMES A. HALL,
Machine-Shop Practice Division
CHARLES W. BEESE,
Management Division
G. E. HAGEMANN,
Materials Handling Division
J. L. WALSH,
National Defense Division
L. H. MORRISON,
Oil and Gas Power Division
W. R. ECKERT,
Petroleum Division
F. M. GIBSON and W. M. KEENAN,
Power Division
WINFIELD S. HUSON,
Printing Machinery Division
MARION B. RICHARDSON,
Railroad Division
JAMES W. COX, JR.,
Textile Division
WM. BRAID WHITE,
Wood Industries Division

Fuels

Coke-Recovery Methods

F-15 A member of the Society wishes to know if any of the readers of MECHANICAL ENGINEERING have attempted to recover coke from ashes, or if they know what progress has been made in this work or in the perfection of equipment to effect the recovery of combustible materials.

Our experience has been confined to the use of the flotation method for separation of clinker from coke, using water of about 1.35 density, secured by dissolving clay. A "Kolumbus" coke separator is used for separation of clinker from coke screened over a $^1/_2$-in. mesh screen. Approximate analysis of coke before separation: ash, 30 per cent; combustible, 70 per cent. After separation: washed coke, 80 per cent; combustible, 20 per cent ash. Refuse: 85 per cent ash, 15 per cent combustible. We have not used separators for reclaiming coke combustible from boiler-plant refuse. It is questionable if a profitable operation would result due to loss of fine carbon in flotation liquid if not screened and if screened, to the loss of carbon in screenings. (Henry O. Loebell, Vice-President and General Manager, Combustion Utilities Corporation, New York, N. Y.)

Stacks for Oil Burners

F-16 A heating boiler with a rating of 6400 sq. ft. of radiation surface requires, according to the makers, a stack 20 in. by 20 in. by 60 ft. This rating is based on the use of free-burning coal (either hard or soft). In transferring to oil as fuel, what changes must be made in the flue dimensions to obtain best results?

The writer has nothing in the form of definite data to submit in response to this question, but will merely offer an expression of the prevailing sentiment among consulting engineers in the vicinity of New York concerning the design of chimneys and treatment of chimney problems in connection with plants where oil burners are used.

In the first place, it should be pointed out that these comments pertain solely to the class of oil-burner installation wherein a motor-driven fan or blower is used to provide definite air injection. These remarks naturally do not apply in the case of the gravity oil burner where chimney draft is just as essential as in the case of the furnace burning solid fuel. Undoubtedly, the view of the situation about to be offered is based on the tendency of the modern mechanically operated oil burner to create forced-draft conditions in the furnace of the boiler or heater operated, which effect is sufficient in some instances to create a measurable pressure therein.

After some years of experience with these prevailing types of oil burners in this market, a great many of the consulting engineers and building designers are coming to feel that with the customary mechanically operated oil burners there is very little need for stack draft other than for its utility in carrying away the waste gases from the burner. In fact, it is very common under conditions of operating tests to see a boiler set up in an open yard and operated very much in excess of its capacity by a mechanically operated oil burner, with no stack connected to the boiler whatever.

This and other similar experiences appear to be leading the consulting engineers of prominence to feel that the outstanding requirement for chimney or stack connections for oil-burner-operated heaters is merely that necessary to carry away the waste gases, for which ordinarily very little attention need be given to the function of creating draft. In numerous instances of buildings erected in this vicinity recently, abbreviated chimney sizes have been devised in view of the use of oil burners, and apparently very satisfactory results have been obtained.

It cannot be stated that such experience is becoming general, as the majority of building designers insist that provisions be made for reverting to coal fuel if any shortage of oil should in the future make this necessary, but the experience developed thereby is very interesting. The writer unfortunately has no data concerning the actual reduction in size which would be necessary, but would feel that this could be approximated very satisfactorily by a rough calculation of the maximum volume of gases that would be discharged by any given oil burner, and then to proportion the chimney or stack so that this volume could exit through it and the flues without the necessity of an unduly high velocity. (C.-W. O., New York, N. Y.)

Machine-Shop Practice

Fitting a Loose Pulley to a Shaft[1]

MS-13 The bore of a pulley is too small to accommodate the shaft upon which it will run loose. Is it recommended that the cold-drawn shaft be turned to fit, or is it better to rebore the pulley and thereby preserve the hardened "skin" of the shafting?

Bore the pulley if it does not harm the bush. If it does, turn the shaft. The "skin" is not enough harder than the center to make much difference. Smoothness is more important than hardness. (Forrest E. Cardullo, Chief Engineer, G. A. Gray Co., Cincinnati, Ohio.)

Management

Selling Price During Slack Periods[1]

MG-5 The busy season of a small tool-manufacturing plant extends from January to May, inclusive. June, July, August, and September are rather slack months. The burden is distributed according to the direct labor, a factor of approximately 160 per cent being used. During the summer months competition from smaller shops in the neighborhood is keen, owing to the small overhead of the smaller shops. Is it advisable to make the selling price low enough to sell the tools even though the books show a loss, according to the factor? If this is not done there will be no summer work and the overhead will be nearly as great.

[1] This subject has been discussed in a previous issue.

An attempted solution of this problem along the lines suggested will lead to ensnaring difficulties. The low summer price will become the all-year price. Reciprocal action on the part of competitors is very likely to make an even lower price level prevail. The questioner will find much sound philosophy on this question in two articles in the March, 1927, number of *Veneers*, published by the S. H. Smith Company, Indianapolis, Ind.—"On Operating to Capacity," page 19, and "Price Stability," page 29. (George A. Prochazka, Jr., Engr., Genl. Mgr., Central Dyestuff & Chemical Co., Newark, N. J.)

Miscellaneous

WELDED MACHINE BASES

M-5 In discussing the subject Arc Welding as Applied to Manufacturing, on page 413 of the May, 1927, issue of MECHANICAL ENGINEERING, J. F. Lincoln makes the statement in referring to lathe beds that "Since such a bed is called upon to withstand a bending stress. . . . . ." Is it not also true that the beams are in torsion, and this condition may greatly complicate matters?

This is quite true. There are conditions where I-beams may not be suitable, and where torsion would require heavier sections than bending.

The strength of an I-beam in torsion is extremely difficult, if not impossible, to calculate. Through the kindness of the mechanical division of Swarthmore College a 3-in., 5.5-lb. I-beam 24 in. long was tested in their torsion machine until it showed marked permanent set. Curves plotted from the information gained showed that there was a very small permanent set from near the origin of the curve. This was no doubt due to the difficulty of making end connections that would fit exactly. It was so small, however, that it did not vitiate the results.

In torsional strength the beam was about equal to a round rod $^{13}/_{16}$ in. in diameter. In torsional rigidity it was about equal to a round rod 1 in. in diameter. It was able to stand 17 times the moment in bending that it could stand in torsion.

The operator of the testing machine expressed his impression from considerable experience in the words, "The I-beam is about as strong in torsion as a piece of $^{3}/_{8}$-in. gas pipe."

If we consider a lathe 10 ft. between headstock and tailstock, driven by a 10-hp. motor turning a piece 3 ft. in diameter and 8 ft. long at 6 r.p.m. (about 60 ft. per min.), a pressure of 5000 lb. will be exerted on the tool. When the tool is in the middle of the 10-ft. span the bending moment, which will be almost all on one I-beam, will be 150,000 in-lb. The torsion moment is carried by both beams if they are well fastened at the ends, and is equal to 5000 × 18 in. or 90,000 in-lb., which is 45,000 in-lb. per beam. As the load in bending is 3.3 times as great while the strength in bending is 17 times as great, it will require over 5 times as strong a beam to resist the torsion.

The writer uses for the section modulus of a non-closed section the formula $^{1}/_{3} \times W \times t^2$, in which W is the sum of the widths of the various parts and t is the average thickness. For the 3-in. beam t is 0.2 in. and W is twice flange width 2.33 (equal to 4.66) plus depth of web 2.7, equal to 7.36 in. In checking the accuracy of this formula with the above tests, all depends on what value is assumed for the yield point in shear for the metal used. The writer finds, however, that the formula is close to the test result. He has previously tested it with thin rectangular shapes, with which it is not so close but on the safe side.

This proves that to get the most economical beam in torsion one will in most cases choose a beam of less height but with a thick web. This is the opposite of practice in bending.

For torsion it would undoubtedly be better to weld two deep channels together to make a box. The above formula does not apply to such a section, and it is suggested that some college test such a beam. (A. T. Kasley, Research Department, Westinghouse Electric & Manufacturing Co., Philadelphia, Pa.)

CURING POROSITY IN ALUMINUM CASTINGS

M-6 What methods are available to cure porosity in aluminum castings?

While dipping in sodium silicate is often used as a cure for porosity in aluminum castings, it is believed that where aluminum castings are to be used in places where tightness is important, the only proper treatment is to throw the porous castings away. (Lieutenant R. S. Barnaby, (CC)U.S.N., Bureau of Aeronautics, Navy Department, Washington, D. C.)

COMPARATIVE WEIGHTS OF MACHINE PARTS

M-7 For a given weight in a machine part subjected to vibration and the usual tension and compression stresses, which material is preferable, a light alloy of the duralumin type or a heat-treated alloy steel?

(*a*) The choice of materials in this case is dependent upon a number of factors, among which may be mentioned the shape and form of member, the maximum stress obtaining, and the ratio of the vibrational stress range to the primary stresses. Comparatively few data are available concerning the last factor, but from a consideration of the other two it may be pointed out that for parts of equal weight and similar in cross-sectional shape (that is, round or square), it is necessary to provide a heat-treated alloy steel with an endurance limit of not less than 70,000 lb. per sq. in., and in the case of a rectangular cross-section of constant width but varying depth (a duralumin part has a depth of 2.84 times the depth of a steel part), it is necessary to provide a steel with an endurance of not less than 120,000 lb. per sq. in. in order that the steel part may be equal in strength to the duralumin part. (Robert L. Streeter, Vice-President United States Aluminum Company, Pittsburgh, Pa.)

(*b*) The following extracts are quoted from Army Air Corps Information Circular, Volume 6, No. 568, dated September 1, 1926. ". . . [For] tension members subjected to much vibration and to stress reversals, both steels are preferable to duralumin." The two steels referred to in the above statement are chrome-molybdenum steel having an ultimate tensile strength of 95,000 lb. per sq. in., a yield point of 60,000 lb. per sq. in., and modulus of elasticity of 28,000,000, and a heat-treated chrome-molybdenum steel having an ultimate tensile strength of 150,000 lb. per sq. in., a yield point of 105,000 lb. per sq. in., and a modulus of elasticity of 29,000,000. "Thus while duralumin has the highest specific strength of the above materials, it is never used for pure tension members and its use near engines where there is much vibration is not favored because of its fatigue limit." The Bureau of Standards has conducted a long series of experiments on the fatigue limit of duralumin, and it is believed that much valuable information along this line can be obtained from that Bureau. (Lieutenant R. S. Barnaby, (CC)U.S.N., Bureau of Aeronautics, Navy Department, Washington, D. C.)

Questions to Which Answers Are Solicited

ALUMINUM ALLOY HARDNESS

M-8 How do aluminum-piston manufacturers obtain a Brinell hardness of 125 to 150 with a 500-kg. load with their regular piston alloy, which is about 90 per cent aluminum, 10 per cent copper, and small quantities of magnesium and iron? What heat treatment is used to obtain this hardness, and what is the exact alloy used?

DEFINITIONS OF "STRESS" AND "STRAIN"

M-9 At a recent meeting of Committee E-1 of the American Society for Testing Materials the definitions of terms relating to methods of testing given on page 1109 of Part I of the Proceedings of 1924 were passed for adoption as standard, subject to the approval of Committee E-8. These definitions involve defining "stress," where used without a qualifying term, as meaning unit value, and similarly as it concerns "strain." This usage is contrary to the inquirer's understanding of general engineering practice where "stress" usually indicates the total quantity, and unit stress or stress per unit of area the unit quantity. An examination of available literature and thorough discussion in this department should prove valuable, and may prevent misunderstandings after the standard has been adopted.

Engineering and Industrial Standardization

Safety at Home

DURING the first four months of 1927, more people were killed right in their own residences than on the streets and highways. Unless accidents in homes are checked, the problem of safeguarding residences will become a more difficult question to solve than the task of protecting the American people, both motorists and pedestrians, while they are on the streets and highways.

Hundreds of communities, national organizations, state bodies, and local associations are trying to cope with the traffic problem, but practically nobody is worried about the national toll taken by home accidents. Organized safety efforts are reducing the number of accidents outside the home. Thousands of industrial shops, factories, and plants and hundreds of motor clubs, safety councils, owners of motor-vehicle fleets, and other bodies are holding meetings to tell people how to avoid accidents while at work and outside their workrooms but there are no rallies or organized efforts to point to the perils that are to be found in every home.

More than 18,500 men, women, and children met accidental deaths in their own homes last year. According to the National Safety Council, which is trying to prevent accidents everywhere, the same causes that are responsible for the toll taken in industry and on the streets are behind the fatality records in homes and the same preventive measures will safeguard residences.

It is really immaterial whether you or a member of your family gets involved in an accident that occurs on the street, at work, or at home. Every accident is accompanied by the same pain, the same medical or funeral expense, the same danger of physicial handicap with its attendant reduction in earning capacity, but the home accident usually lacks any possibility of financial compensation.

Today even the children in public, private, and parochial schools are being taught the A.B.C. of safety. But the American people are not aware of the hazards that exist right in their homes. Here is a situation worthy of the consideration of such organizations as the Visiting Nurses' Association, Associated Charities, Parent-Teachers' Associations, National Federation of Women's Clubs, Salvation Army, Y.M.C.A., Catholic Sisters, Jewish Welfare Boards, and numerous other societies. To correct this evil—and accidents constitute an evil—the public should be informed of the perils confronting them at home. Education is the key to the problem. All of us should know of these dangers and learn how to prevent home accidents.

The National Safety Council is making an effort to call attention to the perils we face daily. For this purpose it has prepared for he daily press an interesting series of questions and answers.

Notes of Progress

FROM time to time we plan to give in this section of MECHANICAL ENGINEERING brief statements covering the progress of the projects for which the A.S.M.E. is sponsor or joint sponsor. The following is such a review prepared the first week of June.

Shafting. The third report of the Sectional Committee on the Standardization of Shafting was presented to the Society for discussion at the Spring Meeting at White Sulphur Springs. It now goes to the Council for approval by letter ballot, following which this proposed standard Code for the Design of Transmission Shafting will be transmitted to the American Engineering Standards Committee for approval as a Tentative American Standard.

Milling Cutters. The large representative Committee on the Standardization of Milling Cutters held a satisfactory meeting at White Sulphur Springs, W. Va., on May 24. The reports of the several Sub-Groups were presented and freely discussed. The sets of standard dimensions which they propose are being completely revised and will soon be distributed for criticism and comment in printer's-proof form.

The proposed American Standard for Shell End Mills is now in that form and available to those who apply to A.S.M.E. Technical Committees Department, 29 West 39th Street, New York.

Machine Tapers. Under the sponsorship for Small Tools and Machine-Tool Elements there has been organized a representative Sub-Committee on the Standardization of Machine Tapers. This Sub-Committee held its first meeting in New Haven, Conn., last September and its second meeting at White Sulphur Springs, W. Va., on May 24. At this second meeting the Committee decided to send out a questionnaire to ascertain what support it might expect to the proposal that the present practice be gradually changed over to the use of one taper, to be known as the American Standard. The suggested rate of taper was $^3/_4$ inch to the foot, or 1:16.

Identification of Piping Systems. The Special Editing Committee has completed its work and the proposed tentative standard is now in the hands of Sectional Committee members for release to the sponsor bodies. During the editing process a condensed statement of the proposed scheme for the identification of piping systems was developed from the reports of the three Sub-Committees. It now covers identification by color, bands, and other markings. A large amount of supplementary material is to be published as appendices.

Printer's proofs have been sent to the technical press for publication and to interested concerns for criticism and comment.

Plain Limit Gages. The special Editing Committee, Earle Buckingham, Chairman, has made good progress recently toward the completion of the second report of the Sectional Committee on the Standardization of Plain Limit Gages for General Engineering Work. This report is entitled, "Methods of Gaging and Specifications for Plain Limit Gages."

Printer's proofs of this proposed standard have been distributed during the past few weeks for criticism and comment, and it is expected that final action will be taken on the report by the Sectional Committee very soon.

Steel Pipe Flanges and Flanged Fittings. The proposed Tentative American Standards for Steel Pipe Flanges and Flanged Fittings designed for steam pressures of 250, 400, 600, 900, and 1350 lb. per sq. in. have been approved by the Heating and Piping Contractors National Association, the Manufacturers Standardization Society of the Valve and Fitting Industry, and The American Society of Mechanical Engineers in their capacity as joint sponsors for this project, and they have in turn transmitted these standards to the A.E.S.C. for approval and designation as Tentative American Standards.

Cast-Iron Flanged Fittings. Letter ballots of the Sectional Committee have been completed on the proposed American Standards for 125-Lb. and 250-Lb. Cast-Iron Flanges and Flanged Fittings. These two reports have accordingly been submitted to the three sponsor organizations for their approval and transmission to the A.E.S.C.

Screwed Fittings. The proposed Tentative American Standards for 150-Lb. Malleable Screwed Fittings, the 125-Lb. Cast-Iron Screwed Fittings, and the 250-Lb. Cast-Iron Screwed Fittings are now in the hands of the sponsor bodies for approval. Subsequently they will be presented to the A.E.S.C. for designation as Tentative American Standards.

Wrench-Head Bolts and Nuts and Wrench Openings. The first report of the Sectional Committee on the Standardization of Bolt, Nut, and Rivet Proportions to reach the A.E.S.C. is that on Wrench-Head Bolts and Nuts and Wrench Openings. This standard has now received the designation "Tentative American Standard," and is available in pamphlet form upon application to the A.S.M.E. Publication Sales Department, 29 West 39th Street, New York, N. Y.

Small Rivets. This proposed standard, which covers ferrous rivets $^7/_{16}$ in. and under, was submitted to the A.E.S.C. for approval as an American Standard on May 7. Within a short time, therefore, it will be available in pamphlet form.

Aeronautical Symbols. A proposed set of letter symbols has been developed by a Sub-Committee of the Sectional Committee on the Standardization of Scientific and Engineering Symbols and Abbreviations. This tentative report is now in type and is being distributed for criticism and comment. For copies apply to the Secretary of the Sectional Committee, Preston S. Millar, Electrical Testing Laboratories, 80th St. and East End Ave., New York, N. Y.

Transmission Chains and Sprockets. The organization meeting of this Sectional Committee under the procedure of the A.E.S.C. was held on December 10, 1926, at which F. V. Hetzel was elected temporary chairman. Sub-Committees on the Standardization of Roller Chains and Silent Chain were appointed at that time. A meeting of the Committee on Roller Chains is planned for the near future.

Drawings and Drafting-Room Practice. In March, April, and May the following four Sub-Committees of the Sectional Committee on Standards for Drawings and Drafting-Room Practice held their organization meetings in New York: Layout, Line Work, Specifications for Paper and Cloth, and Lettering. Within a few weeks, therefore, tentative proposals drafted by these Sub-Committees will be ready for distribution for comment and criticism.

Code for Pressure Piping. The Sectional Committee in charge of this important piece of work has divided itself into the following Sub-Committees: Plan and Scope, Power Piping, Hydraulic Piping, Gas and Air Piping, Refrigeration Piping, Oil Piping, Piping Materials, and Fabrication Details. Recently Chairman E. B. Ricketts and the members of the Plan and Scope Committee held a conference with the chairmen of the several sub-committees, the purpose of which was to define as near as possible the scope of each sub-committee's work. It is expected that considerable progress in the development of this code will be made during the coming months.

Graphic Presentation. This Sectional Committee held its organization meeting on December 3, and has since held meetings of its Sub-Committees on Terminology and Survey of Current Practice. The scope of the project has been divided into the following subdivisions, each assigned to a Sub-Committee: Plan and Scope, Terminology, Time Series Charts, Non-Time Series Charts, Survey of Current Practice, and Engineering and Scientific Graphs. Drafts of the proposals of these sub-committees will no doubt be completed within a short time.

Recent Papers of Interest

THE following four articles on Standardization in its various aspects have appeared in magazines during the past year or so. Reprints of some of them are available on application to the A.E.S.C. office, 29 West 39th Street, New York.

"SOME ENGINEERING ASPECTS OF STANDARDIZATION," by Emil G. Bern, *General Electric Review*, October, 1926. Reprints available.

"ENGINEERING AND INDUSTRIAL STANDARDIZATION," by Staff Writer, MECHANICAL ENGINEERING, February, 1927. Reprints available.

"STANDARDIZATION AS A COST REDUCER," by B. M. Swarr, Director, Standards Section, General Purchasing Committee, General Motors Corporation. Published in August 26, 1926, issue of *American Machinist*. Reprints available.

"CONSUMERS IN WONDERLAND." A series of articles on "What We Get for Our Money," by Stuart Chase and F. J. Schlink, published in six succeeding issues of *The New Republic:* Feb. 2, 1927, Introductory Statement; Feb. 9, The Twin Gods of Bad Business: Adulteration and Misrepresentation; Feb. 16, What We Get for Our Money When We Buy from Quacks and Venders of Cure-Alls; Feb. 23, How Standardization Helps and Hinders the Buyer; March 2, In Which The Authors Give Some Suggestions as to the Way Out of Wonderland; and March 9, Responses from Wonderland.

A Correction

ON page 622 of the June issue of MECHANICAL ENGINEERING, the first equation in Mr. Beresi's discussion of Mr. Frame's paper on Stresses Occurring in the Walls of an Elliptical Tank Subjected to Low Internal Pressures should have been written

$$p_1 = \frac{ab\pi p}{\text{cross-sect.}} = 725 \text{ lb. per sq. in.}$$

A.S.M.E. Boiler Code Committee Work

THE Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, Mr. C. W. Obert, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of the Society for approval, after which it is issued to the inquirer and simultaneously published in MECHANICAL ENGINEERING.

Below are given records of the interpretations of the Committee in Cases Nos. 545 and 547, as formulated at the meeting of April 15, 1927, all having been approved by the Council. In accordance with established practice, names of inquirers have been omitted.

CASE NO. 545

Inquiry: Is it permissible, under the requirements of Par. P-268, to reinforce the material at openings in boilers to provide the specified number of threads, by swedging or drawing the metal so as to substantially upset or increase the thickness around the opening?

Reply: It is the opinion of the Committee that expanding or upsetting the metal around an opening to cause it to be thickened to meet the requirements of Par. P-268, is permissible for tappings not exceeding $^3/_4$ in. pipe size.

CASE NO. 547

(In the hands of the committee)

New Iron Process

MUCH interest has been aroused in iron and steel producing circles by the announcement of a low-temperature process for the production of pure iron in granular form from ore at one operation. The main stages of the process, which has been devised by Thomas Rowlands, a Sheffield steel manufacturer, are distillation of coal, preliminary reduction of iron ore, complete reduction of iron ore, magnetic separation of pure metallic iron, revival of spent gases. Dr. Percy Langmuir, to whom the process has been submitted, considers that it holds much promise to the maker of carbon and alloy steels. It will not, he says, in any sense compete with existing blast-furnace methods for the production of pig-iron, but will fill a special field of its own for the supply of pure metallic iron as a foundation material for high-class steel-making. Dr. Frank Rogers is of opinion that it embodies the correct principles and applies them essentially correct in detail. In describing it he states that iron ore in suitable form, usually crushed, is charged continuously through a chamber, where it is mixed with the reducing agent, usually a special form of coke, and treated with reducing gases. The coke and the gases are simultaneously produced in another section of the apparatus. The reduced iron passes forward through a cooling chamber, and after being discharged, is separated magnetically from the dross. Precautions are taken for the conservation of heat, with the result that the coal consumption per ton of metal produced is something like one-third of that which is necessary to produce one ton of pig-iron. The granular iron obtained forms a raw material for the production of steel, and as it is of a purer character than pig-iron its cost, in view of the economy in fuel and for other reasons, is comparatively low. It should be decidedly cheaper than some of the more expensive raw materials at present used in steel-making and, consequently, it is thought, will be of most importance in connection with the most expensive grades of steel.—*The Times Trade and Engineering Supplement*, June 4, 1927, p. 277.

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and Attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities, Papers and Proceedings of

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An Earnest of the Future

THE latter part of May and the first week of June will go down in the history of transportation as having witnessed three remarkable flights. Practically at the same time that Charles Lindbergh flew from New York to Paris in a Ryan monoplane with a Wright Whirlwind air-cooled motor, Carr and Gillman attempted to fly from London to Karachi, India, in a Hawker-Horsley plane with a 700-hp. Rolls-Royce Condor engine but landed about 200 miles short of their goal. Measurements would indicate that to Lindbergh belongs the palm for the longer flight, he having exceeded the British fliers by from 100 to 200 miles according to different estimates. So rapid is the advance of aviation, however, that Lindbergh's own record stood less than a fortnight, Clarence Chamberlin, accompanied by Charles Levine as a passenger, having flown from New York to Eiseleben, Germany, within 100 miles from Berlin, and covered a total distance of close to, if not in excess of, 4000 miles. (However, it must be remembered in this connection that Chamberlin flew until he consumed his last drop of gasoline, while Lindbergh set Paris as the goal, and when he landed he still had quite a little fuel left in his tanks.)

It would be superflous to emphasize by referring to these flights the comparatively high state of development which aeronautics has reached today. Of interest, though, is the fact that the flights by Lindbergh and Chamberlin show the state of development which this new method of transportation has already reached in America. The Ryan and Bellanca planes are of a type entirely developed in this country, differing in many respects from any of the European planes. The Wright Whirlwind motor is likewise a strictly American development, and not in any way a copy or adaptation of European motors. The efficiency shown by the Bellanca plane in its 51-hr. endurance test flight over Long Island and in the New York-to-Germany flight, as well as the success of Lindbergh in his flight across the northeastern tip of the American continent and the wastes of the Atlantic, bear witness to the fact that at least from an engineering point of view, aeronautics is by no means as backward in its development in the United States as some would have it inferred.

At the same time, these two flights have also shown the existing limitations of aircraft as a means of long-distance high-speed transportation. Lindbergh flew alone, and the glorious reception which he received may be taken as recognition that very unusual courage and skill were required to accomplish this feat. In the Chamberlin flight a passenger was carried, whose wife duly fainted when she saw her spouse wing off on his perilous trip.

In either case, the first reaction to the flight is that it is a very unusual and dangerous undertaking—useful because of its future implications but not because of its present commercial value. It should be remembered, however, that in transportation the stunt of today often becomes a commonplace of tomorrow; for example, in automobile development feats on the race track have taught engineers how to achieve similar performance on the highways.

The question may be asked, What is going to be the next step? A careful analysis of the efficiency of the engine and its fuel consumption would indicate that for planes such as the Ryan, Bellanca, and Hawker, a flight of 5000 miles is practically the limit under the most favorable conditions. It should be remembered, however, that the fuel efficiency of the best airplane engine of today is still under 20 per cent, as compared with the 35 per cent of the Diesel engine. Furthermore, the same engine, with certain changes which are already in sight, could develop a greater efficiency with modified fuels, such as special cycle hydrocarbons, and under certain conditions, chemical dopes. Aerodynamic laboratory work has also shown the possibility of producing planes of greatly reduced resistance combined with increased lifting power. It would therefore be no exaggeration to state that within a comparatively short time, say, within the next year or two, non-stop flights from New York to Siberia or New York to Tokyo, and perhaps flights around the world in, say, three jumps will become possible. For various reasons it is far less clear when the fortunate time will come in which an ordinary business man who is willing to pay a somewhat greater price will be whisked from New York to Southampton in a day with the same certainty with which he can now make the passage between these two points in the space of five days.

Few, however, who have followed closely the development of aviation will now doubt that the time is coming when a man will be able to breakfast in New York and keep a luncheon engagement the next day in London. As a matter of fact, he is already able to make such an appointment by telephone, and all that is needed is better planes, a suitable organization of transport, and refueling and relief stations in the ocean to permit of his keeping it.

Management—An International Problem

ONE of the important factors in the industrial well-being of the United States is the intensive development of mass-production methods in which mechanical engineers have played a heavy role. As we survey the situation today, it is quite obvious that the United States is essentially a mass-production country. Its broad expanse, its great natural wealth of minerals and rich lands which support its enormous population, are the basic elements that make specialization and high-speed production possible. The United States cannot take the credit for first thinking of mass production but she can take the credit for successfully using its advantages for the well-being of her people.

Mass-production and the coördinate reasons for American industrial success are, therefore, the subject of careful scrutiny. Foreign missions, large and small, official and otherwise, pour into this country to study its methods. American engineers, economists, and commercial leaders are drafted into an expeditionary force of rehabilitation for foreign industries. International congresses on economics, commerce, and management deliberate on the advantages and disadvantages of American methods. In the development of mass production and the qualities of thought which must accompany it, the United States has, therefore, made a great contribution to world knowledge.

The conditions surrounding very few other countries are such that mass-production principles with the collateral arts of mass marketing and mass distribution can be applied exactly as they are in the United States. But other countries meet their problems of improving the welfare of their peoples by methods peculiar to their own conditions and productive of the best results under those con-

ditions. A complete statement of the successful solution of all management problems must, therefore, include a contribution from each country, and a well-rounded impression of complete management technique must come from a clearing house of information open to all countries. Such a clearing house is to be held this fall in Rome in which the countries of Europe will participate. An active American committee is engaged in preparing a program of papers presenting actual cases of American success in solving management problems.

Research in the Steel Industry

A FEW years ago on the award of the Bessemer Medal to a great American metallurgist one of the trade papers took occasion to remark that the few awards of this great honor to Americans constituted proof that in this country where tonnage was supreme, quality and improved markets were losing out. As a basic industry, steel has had a continuing relation with the influence of science on mechanical design. A portion of the industry has realized the benefits that have come from research in alloys. The automobile, the railroads, the increased usage of pressed metal, the development of welding, have been factors in securing new materials or changed varieties of old ones as a result of scientific investigation. The great body of the steel industry had advanced ponderously but apparently oblivious of the warning of the prophets that the industry which fails to employ science is doomed to destruction.

All of this being so, the recent announcement of the Steel Corporation that a comprehensive research program has been instituted under competent scientific guidance is most welcome. Dr. John Johnston, Professor of Chemistry at Yale University, is to head the new Department of Research and Technology with the title of Director. He will have the aid of an executive council of four which will include Dr. R. A. Millikan, of the Norman Bridge Laboratory of Physics at Pasadena, Calif. The new department will be responsible solely to the Finance Committee of the Corporation. It is the announced plan to draft seasoned employees of subsidiary companies to start the scientific work.

The initiation of this new influence in the steel industry under such favorable auspices and with strong assurances of ample financial support is most encouraging. Mechanical engineers will welcome improvements in materials that permit progress in mechanical design.

Flood Relief

THE Mississippi is returning from its flood debauch. Excitement prevails over the steps to be taken to prevent a similar disaster in the future. Noisy demands for great expenditures and fantastic proposals of quick cures are rampant. As yet, however, there has been no action on the appointment of an engineering commission to examine into and report upon a comprehensive plan for Mississippi control which the American Engineering Council advocated.

The situation is somewhat analogous to that prevailing in aviation when Colonel Mitchell's storm broke. The result was a Morrow Board Report whose recommendations appeared promptly in legislation with the result that now aviation is well along its path.

The Mississippi River has unleashed a great flood. Its effect on lives, property, and the orderly existence of man, permits a thrilling dramatization. The engineering profession and the public want the assurance that would come from dispassionate investigation by qualified men. The Mississippi cure needs a study comparable to that leading up to the Morrow Report.

Mechanical Engineers at a Resort

SEVENTEEN years is too long a period to elapse between A.S.M.E. meetings at resorts. Those who were at White Sulphur Springs for the Spring Meeting unanimously supported that view. Every one enjoyed the entertainment, the sessions, and the chances to "swap" experiences. Lights burned late in the cottages as self-determined groups settled Society policies and replayed their golf games. They had a good time and thus badly jarred the tradition that engineers are technical dullards who do not know how to enjoy themselves.

Atlantic City—1910; White Sulphur Springs—1927. The span is too long. What is the right interval? There are enough members of the A.S.M.E. to make an annual resort gathering a success. The interest would build up from year to year and gradually an event comparable to the Annual Meeting in compelling interest would result. What are the bad features of an annual resort meeting?

The Place of the A.S.M.E. in Research

ENGINEERING Research has many aspects. First, there is that in which prompt and direct returns for the money and time invested are desired. Such researches or investigations are best made for and under the direction of individuals or single firms.

Second, groups of individuals and firms may join their financial and engineering resources and take up a study which is broad enough to interest all of the supporting group in one way or another. By this method problems of a somewhat general character can be taken up but there is still direct supervision and fairly rapid dissemination of the information obtained.

Third, for the solution of many problems in the field of engineering as well as in the field of medicine, large endowments are necessary to insure adequate facilities, trained personnel, and the assurance of continuity of experimentation.

In its research activities the Society has passed through the first stage, is now functioning in the second, and many believe it should move on into the third.

Since its organization the A.S.M.E. has encouraged its members to study their individual problems in a systematic way and, when appropriate, has urged the presentation of the results in the form of papers for discussion and record. Shortly after the organization of the A.S.M.E. Main Research Committee, group action was added to this program. Special Research Committees were formed on specific subjects which were of interest to groups of the Society's members.

The financing of the first type of research is relatively simple and the support is generally in direct proportion to the information received or desired. The group method permits firms to have a part in the investigation without investing large sums for facilities and workers. The $32,000 contributed by the Society for research from the year 1919–20 to date has been applied to the formation and development of these group researches. A relatively small part of that money has covered the actual expense of the research. On the contrary, the yearly appropriations of from $3000 to $14,000 have been used to encourage the contribution of many times these amounts by interested firms and individuals. The Treasurer's records show that an investment by the Society of $32,000 in the last seven years has resulted in the collection by its various Special Research Committees of approximately $142,000 for research in the field of mechanical engineering. By this method the following funds have been created; Steam Tables, $65,000; Boiler Furnace Refractories, $18,000; Elevators, $32,000; Fluid Me'ers, $1600; Lubrication, $800; Mechanical Springs, $4500; Bearing Metals (pledged), $14,000; Gears, $6500; while fourteen other, Special Research Committees of the Society are in the various stages of development leading up to the creation of research funds to support their activities. The rules of the A.S.M.E. Main Research Committee provided that after a project has been approved by the Council a special committee to develop and supervise the work shall be formed. This special committee drafts its own scope, plans its campaign for funds, and controls the expenditure of those funds. The Main Committee exercises only general supervising and stimulating powers.

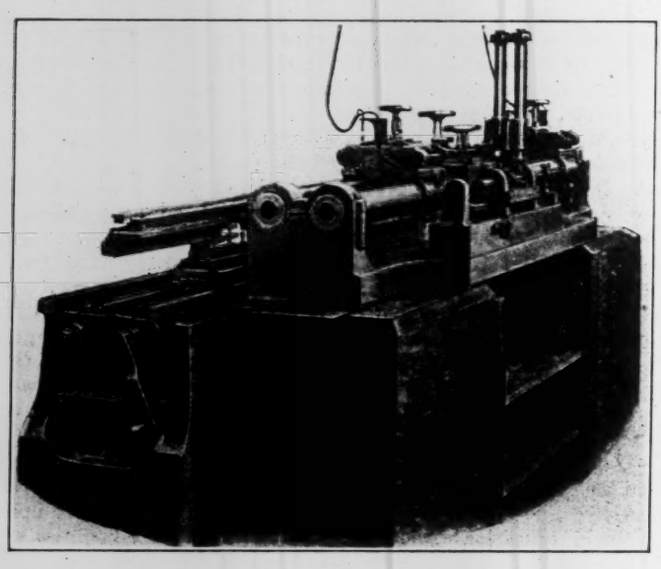
Experience has shown that this type of organization and procedure coupled with the deposit of the funds in the treasury of the Society is generally acceptable to the mechanical industries. There is, however, a limit to the amount and kind of research which can be conducted in this way. The A.S.M.E. Main Research Committee is accordingly giving this subject very careful study and expects to present an enlarged plan for discussion at the Annual Meeting next December.

The Quest for the Basic Inch

SOME very interesting points regarding the pursuit of that elusive germ of American standardization—the basic inch—have come to light through a reference given by W. H. Weingar.

In delving into the archives of the Pratt & Whitney Co., Mr. Weingar came across a book entitled Standards of Length and Their Practical Application, published at Hartford, Conn., in 1887. This book is particularly interesting in that it describes pioneer work done more than forty years ago by George M. Bond, who is still an active member of The American Society of Mechanical Engineers. Furthermore, it brings out the active role which the Society was playing at that early day in encouragement of such pioneers as Mr. Bond in the foundation of authentic basic standards, and in the technique of accurate gage making. The Society thereby gave distinct impetus to the higher development of the interchangeable system of manufacture upon which American industrial leadership in so large a measure depends.

The book was not only edited by Mr. Bond but also very important portions of it were written by him and describe his own work. The opening chapter is a lengthy report by William A. Rogers, professor of Astronomy at Harvard. By a fortunate chance it had been possible to divest Professor Rogers' mathematical and scientific talents from the consideration of millions of miles to millionths of inches—for a time.



THE ROGERS-BOND UNIVERSAL COMPARATOR

Professor Rogers' report covers the exceedingly scientific and painstaking investigations of the exact relations existing between certain commercial standards of length used in controlling gage manufacture and several official standards such as the Imperial Yard of Great Britain, Bronz Eleven of the United States Bureau of Weights and Measures and the Metre des Archives of France. Professor Rogers' investigations determined within such close limits as "25 millionths of an inch" the relations between the official and the commercial standards.

Following Professor Rogers' report there is a report of the A.S.M.E. Committee on Standards and Gages. This is mainly concerned with the Rogers-Bond Comparator, an ingenious instrument of extreme precision which was designed by Professor Rogers and Mr. Bond and constructed under their supervision by the Pratt & Whitney Company. The instrument was designed to fulfil three purposes: first, to compare line-measures of length with attested copies of standard Government bars; second, to subdivide these line measures into their aliquot parts and to determine the errors of such subdivisions; and third, to reduce these line-measures to end-measures for practical use in shops.

This Report was read at the Annual Meeting in New York, November, 1882. It bears the signatures of J. Sellers Bancroft, Henry Morton, S. W. Robinson, Oberlin Smith, E. H. Parks, Ambrose Swasey, Charles T. Porter, Alfred Betts, and George R. Stetson.

In the Rogers-Bond Comparator two microscopes, with magnification of 150 diameters, were employed for reading divisions. These had micrometer eyepieces, the divisions of which represented $^1/_{50,000}$ of an inch. In reading, it was possible to sub-divide these divisions by eye into tenths.

The microscopes were mounted upon a carriage which slid freely upon two cylindrical guides which were heavy tubes of hardened tool steel made cylindrical and straight by grinding. The guides were provided with counterweighted levers to overcome flexure from their own weight and that of carriage and stops. The stops served to limit the motion of the microscope carriage and held the carriage in unvarying contact by electro-magnets.

The Comparator was mounted upon two separate sets of foundations—one set of stone-capped brick piers for the body of the machine and a separate set for the movable table. This table had rapid lateral and vertical adjustments, its traverse parallel to the microscope carriage being about forty-five inches. It also had fine adjustments for parallelism, focus and position. It would accommodate several standard bars at once and thereby avoided temperature changes due to handling. The opposite side of the comparator had a projecting ledge upon which this table could also be placed when desirable but where ordinarily the caliper device was placed for reducing from line to end measure.

The microscopes were provided with Tolles prisms by which light was turned down through their barrels and illuminated the object under observation directly from above. This made it possible to see both the edges and the bottom of a scratch and to strike the center of ruled lines on plugs set below the surface of bars.

The operation of the Comparator has been described by Mr. Bond as being based upon the theorem that "Two things equal to the same thing are equal to each other." There were five ways of comparing line-measure standards; first, by referring them to a fixed distance between the two stops; second, by bringing defining lines under two fixed microscopes separated about the distance represented by the length of the compared standards; third, by placing two standards side by side, placing a microscope over each, and moving the carriage the length of the bars; fourth, by placing one bar on one side of the carriage and one on the other, a microscope being used on each, and reversing the position of the bars to determine a mean difference; fifth, with the two microscopes horizontal and at a fixed distance apart, as was the condition under which the work was done at the United States Coast Survey Department in Washington.

Curvature of a bar was detected by focusing first upon a tray of mercury and then upon the bar, the micrometer eyepieces enabling the amount of curvature to be accurately determined.

The caliper mechanism by which line-measure was reduced to end-measure consisted of a fixed stop and a movable cylindrical plunger sliding in well fitting bearings to the extent of six inches.

The plunger was pressed against its fixed stop or against anything between the two surfaces by a rod compressing a spiral spring within the plunger.

The caliper was used with two microscopes, one over a finely ruled standard bar, and the other over a finely ruled plate attached to the plunger. First, the faces of the caliper and stop were brought in contact by the rod and spring and one microscope set to bisect a line upon the plunger plate. The other microscope was set to bisect the initial line upon the standard ruled bar, both microscopes being firmly attached to their carriage. The plunger was then drawn back, the piece to be tested placed between the faces of the caliper and stop, and the plunger forced up till the spiral spring was compressed about as before. The carriage was moved until the line on the plate was again bisected by its microscope, and the microscope over the standard bar by the aid of a micrometer determined the exact length of the piece in terms of the subdivisions of the ruled standard. This operation could be performed rapidly and with uniform results. The Comparator is described as having been capable of measuring within $^1/_{150,000}$ of an inch at the hands of Mr. Bond.

The United States Patent Office

By L. W. WALLACE,[1] WASHINGTON, D. C.

IN OCTOBER, 1924, the Committee on Patent Office Procedure appointed by Mr. Hubert Work, Secretary of the Department of the Interior, began to function. The Secretary, in a statement to the committee, said: "A very useful purpose may be served by this committee of prominent patent attorneys, representatives of industrial organizations, and the engineering profession, in making a thorough review of the Patent Office, with a view of simplifying the methods of procedure and expediting the handling of applications for patents."

When the Patent Office was transferred to the Department of Commerce in 1925, the Secretary of Commerce added two engineers to the committee and instructed the committee to continue its work. In April, 1926, a searching, comprehensive, and constructive report was submitted to Mr. Herbert Hoover, Secretary of Commerce. This report represented a great deal of painstaking work on the part of the members of the Committee and that of a number of engineers loaned to the Committee by various agencies as staff men. A sub-committee composed of management engineers, assisted by the engineering staff, made a thorough engineering study of the equipment, layout, and internal procedure of the Patent Office. A corresponding analysis was made of the legal and legislative phases of the situation by sub-committees largely composed of attorneys. These sub-committees consulted with many attorneys in public and private life. The resulting report contained 108 specific recommendations designed to facilitate and improve the work of the Patent Office, and to simplify and to clarify the legal aspects of the patent system.

Few reports have been so widely accepted; few have fallen into such sympathetic hands and immediate application. The Hon. Thomas E. Robertson, Commissioner of Patents, and his associates, as soon as the report was placed in their hands, earnestly and thoughtfully began to apply the recommendations. As a consequence, remarkable progress has been made in many and varied directions. It is the purpose of this article to enumerate some of the major steps of progress, having in mind that such have been almost entirely due to the energy, enthusiasm, and farsightedness of Commissioner Robertson and his associates.

New Quarters Provided

The basic and fundamental recommendation of the Committee was that the Patent Office above all things required a modern, adequate, and properly designed new building and equipment. It was realized that marked improvement in the service rendered to American agriculture, industry, and commerce would not be attained under the housing and equipment conditions now obtaining. It has been definitely decided that an entire wing of the new Department of Commerce Building will be allotted to the Patent Office. In these new quarters the Patent Office will have more square feet of floor space than now. Not only this, but the area will be much better adapted to the purpose than is the case in the present building. Until the new building is ready for occupancy, additional space will be obtained in other buildings adjacent to the Patent Office Building.

By virtue of the Department of the Interior's releasing some of the space it was using in the Patent Office Building, a rearrangement of the filing cases has been made possible. This rearrangement with the addition of a number of new steel filing cases has greatly simplified the storing of patent prints. Not only this, but they are better protected from fire hazard, and more easily assorted and placed in and withdrawn from the files, thus facilitating the work of the office.

It was recommended that the present method of filing material in the examiners' rooms, be replaced by vertical steel filing cases. A new design of vertical steel files is being given a thorough trial.

It was found that the filing wrappers were not giving satisfactory service. In coöperation with the Bureau of Standards the Patent Office has developed a much more suitable and satisfactory file wrapper.

[1] Executive Secretary, American Engineering Council. Mem. A.S.M.E.

Helpful Legislation Relating to the Patent System

In compliance with recommendations of the Committee at the last session of Congress, the statutes were revised in a number of important directions. It has been said by those competent to judge that the revisions made were more constructive and helpful than any legislation passed in the last quarter of a century relating to the patent system. Among the revisions authorized by Congress were the following: The filing fee has been increased by $1.00 for each claim filed in excess of twenty, and the final fee has been increased by $1.00 for each claim allowed in excess of twenty.

The Committee found that the privilege of submitting an unlimited number of claims was being grossly abused. Many applications contained far more claims than necessary, it not being unusual to find over one hundred claims. This practice enormously increased the work of the office and resulted in no particular benefit to any one. The Committee endeavored to limit such an abuse by charging a nominal amount for each claim applied for and each claim granted above twenty. Incidentally this will add considerable revenue, which will enable the employment of more examiners—a thing badly needed.

To expedite the work, to decrease the inequities, and to contribute to the general usefulness of the patent system, the Committee made the following recommendations, which have been enacted into law:

1 That applications must be amended within six months instead of one year, and renewals must be made within one year instead of two. That the length of time allowed for appeals be reduced from one year to six months.

2 Appeals weigh heavily in point of time and cost. Therefore one appeal within the Patent Office has been abolished so that an appeal from the Examiners or from the Examiner of Interferences goes directly to a Board of Appeals consisting of the Commissioner, two Assistant Commissioners, and two Examiners-in-Chief, three of whom shall constitute a quorum. A sixth member of the Examiners-in-Chief has been supplied. The foregoing makes it possible for two Boards of Appeal to sit at the same time. This, with the elimination of one appeal in the Patent Office, will greatly facilitate the handling of appeal cases.

3 Heretofore appeals outside of the Patent Office could be made to the Court of Appeals, and a bill in equity could afterward be filed under Section 4915 of the Revised Statutes. Now either of these courses may be followed, but not both. This means that only one appeal can now be made outside of the Patent Office. This provision will reduce the costs of and expedite patent litigation.

Improved Service Through Revision of Office Rules

Improved service is being rendered to patentees and patent lawyers by the following rules which have been established within the office:

All patents in the Public Search Room are being checked and missing patents placed in the files at a cost of about $20,000. Formerly there was not a complete set of patents on file.

It is now provided that claims shall be renumbered at the time of allowance instead of each time a change is made. This greatly simplifies correspondence and record keeping and reduces liability of error.

Photostat copies of the drawings in each application are made and placed on file at the time the application is received. This serves as a protection against loss and is otherwise beneficial.

It is now permitted to use original drawings in reissue applications. This eliminates the expense of making a new set of drawings and is also a protection against error.

A sufficient number of hard copies for trademarks have been provided and facilities have been increased so that orders for soft copies of patents and trademarks are delivered the same day they are received, and not later than the second day. Copies are promptly supplied the examiners and all delays avoided. It was

found that considerable delay was being experienced, which situation the foregoing arrangement will correct.

OTHER IMPROVEMENTS EFFECTED

Increased appropriations have made it possible to:

Greatly improve the service of the mail room so that delays from that source are now very materially reduced.

Junior Assistant and Associate Examiners are now paid the average of their grade, and an increase in salary will be available for present examiners after July 1. One of the great difficulties is the large labor turnover among the examiners. This is largely due to the low salaries paid for the high type of men required. One of the outstanding needs of the Patent Office is a marked increase in the salaries paid the executive and examiner corps. The Committee made such a recommendation.

According to the Commissioner of Patents, other important improvements recommended by the Committee on Patent Office Procedure have been effected. Among them are the following:

The Issue Division has been *completely* reorganized to change the flow of work and the arrangement of files.

The Trademark Division has been wholly reorganized to carry out the recommendation with respect to the routing of the work.

It was found that in many instances the divisions were not laid out and the equipment placed to conform to the natural and most practicable flow of the work. Several of the divisions were specially studied from this point of view and recommendations made as to the changes necessary. It is gratifying that such changes are being made in so far as present conditions will permit. It is anticipated that this feature will be adequately provided for in the new building.

The usual preamble of printed patents has been omitted, saving from $8000 to $10,000 a year.

Assignments are recorded by the photostat method, saving a number of clerks and reducing the time for recording from 52 to 15 days.

The card indexes for assignments have been completed and have proved the biggest item of saving accomplished.

Adequate stenographic service has been provided for all examiners. This is important as a great deal of time was being spent by the examiners in writing their decisions in long-hand.

The work of the Application Division has been reorganized so as to proceed with the work in small lots in a steady flow. Formerly the lots were going through in large batches, which resulted in alternately a peak and a trough of work—and consequently in inefficiency.

It is apparent from the foregoing brief discussion that many changes are taking place in the Patent Office, all of which means that it is now rendering and will continue to render an increasingly better quantity and quality of service. An instrument of such value to American agriculture, commerce, and industry requires a continuing thoughtful, sympathetic, and active support. There are many other things needing correction before the Patent Office will be able to meet the legitimate demands being made upon it. In a large degree a full measure of accomplishment cannot be realized through the devoted and intelligent efforts of the Commissioner and his staff alone. The changes required must come through the action, interest, and work of those in agriculture, commerce, and industry, who are vitally interested in, concerned with and affected by the Patent system. The Commissioner and his associates have the will to do, as exemplified by their work and as manifested by the following expression which appeared in the Commissioner's report to the Secretary of Commerce, under date of June 30, 1926:

> The Report of the Secretary's Committee is being studied to make use of various other suggestions contained in it to increase the efficiency of the office. In fact, the report will be of immense value. At this time I desire to express the appreciation of the officers of this bureau to the committee, which was composed of men of wide experience in patent, engineering, and industrial problems. Although its members were busy men of affairs, they came again and again to Washington and gave generously of their time and made a most exhaustive study of the Patent Office and its needs. I recommend that the report be printed.

Book Reviews and Library Notes

THE Library is a coöperative activity of the A.S.C.E., the A.I.M.E., the A.S.M.E. and the A.I.E.E. It is administered by the United Engineering Society as a public reference library of engineering and the allied sciences. It contains 150,000 volumes and pamphlets and receives currently most of the important periodicals in its field. It is housed in the Engineering Societies Building, 29 West 39th St., New York, N. Y. In order to place its resources at the disposal of those unable to visit it in person, the Library is prepared to furnish lists of references on engineering subjects, copies of translations of articles, and similar assistance. Charges sufficient to cover the cost of this work are made.

The Library maintains a collection of modern technical books which may be rented by members residing in North America. A rental of five cents a day, plus transportation, is charged. In asking for information, letters should be made as definite as possible, so that the investigator may understand clearly what is desired.

Endurance of Metals Under Repeated Stress

MANUAL OF THE ENDURANCE OF METALS UNDER REPEATED STRESS. A Summary of Views and Test Data with Instructions for Use. By H. F. Moore. Engineering Foundation, New York, 1927. Cloth, 5 × 7¼ in., 63 pp., illus., diagrams, tables, $1.

WITH the increasing use of machines during the 19th and 20th centuries, sudden breaks of parts subjected to many repetitions of stress became more common. Crankshafts, car axles, springs, and many other parts were affected. What caused these apparently sudden breaks in good metal, often disastrous? Could they be prevented? Advent of automobiles, airplanes, turbines, and other high-speed machines increased the need for knowledge of this kind of failure, to which had been given the name "fatigue of metals."

Experimental knowledge and theories of two generations ago were outgrown. New conditions and new materials changed the situation. Developments during and following the World War emphasized the need for more research. Within the past decade a number of important investigations have been carried on in this and other countries. Much remains to be done in the subject, but sufficient new knowledge has been gotten to justify its compilation into a form convenient for users in the every-day work of engineering and industry.

Among the large research enterprises in fatigue of metals has been the one at University of Illinois under the auspices of National Research Council and Engineering Foundation with the coöperation of important industries. Dr. H. F. Moore, resigned professor of engineering materials at that institution, has been in charge. To meet a demand for information in form usable by designing, inspecting, testing, and operating engineers, Engineering Foundation asked Professor Moore to compile a small manual and invited other leading American investigators to coöperate. The product is the manual under consideration, which presents the condensed cream of the knowledge gotten by researches occupying several years of work by a score or more of investigators and many advisers, and costing in the aggregate a several thousand dollars.

This valuable little work can be obtained from Engineering Foundation, at 29 West 39th Street, New York, New York. It

is put forth as a contribution to safety of life and property, and to the welfare of industry and engineering.

ALFRED D. FLINN.[1]

Veneers and Plywood

VENEERS AND PLYWOOD—Their Craftsmanship and Artistry, Modern Production Methods, and Present-Day Utility. By E. Vernon Knight and Meinrad Wulpi. First Edition. The Ronald Press Company, New York, 1927. Cloth, 6 × 9 in., 372 pp., illus., standard grade rules, bibliography, etc., $6.

THE modern method of veneering consists of three, five, or more layers in balanced construction, with the direction of the wood grain in adjacent layers at right angles, as contrasted with the earlier methods dating back to the Pharaohs in which a thin sheet was glued to a thick base without regard to grain direction and frequently not well balanced. This modern product is more frequently called "plywood," a term that has the full sanction of government usage for airplane- and ship-construction purposes, but has as yet found its way into the dictionaries and encyclopedias. Plywood, when properly designed, is a scientifically balanced product, and was recognized by government authority as stronger than steel for airplane purposes, when measured on a weight comparison.

This present publication gives an illustrated outline of the historic uses of veneer, and a most complete description of modern methods of cutting veneer and assembling the thin sheets of wood into properly designed and balanced plywood. It is a valuable reference book for those who want to know the truth about the best modern methods of building veneer or plywood into furniture.

It is fully illustrated, and has many tables and charts, as well as a complete bibliography.

THOMAS D. PERRY.[2]

Standards Year Book 1927

STANDARDS YEAR BOOK, 1927. Compiled by the National Bureau of Standards. United States Government Printing Office, Washington, D. C., 1927. Cloth, 6 × 9 in., 392 pp., 39 figs., $1.

"STANDARDIZATION is the outstanding note of this century." So begins the Standards Year Book for 1927. To justify such a statement the book then proceeds to lay before the reader such an array of general and specific information relating to standardization that one who but scans its pages, however, sceptical of the truth of the startling opening sentence, reaches its end with a realization that standardization has gained a tremendous momentum throughout the world.

To one but little in touch with the forces at work in this particular field, the book opens a surprising view. To those who are actively engaged in the work it presents a compendium of information that will be found useful at frequent intervals.

The title of the volume might with equal propriety be "Standardization Handbook," since it contains such a large proportion of all the information available in print regarding this great "movement which has spread throughout the world with astonishing vitality during the 25 years that have elapsed since the establishment of the National Bureau of Standards."

The twenty-nine fundamental and working standards of measurement which are maintained by the Bureau are briefly described and illustrated. The accuracy of some of these standards is difficult of comprehension.

The outline of the activities of the Federal standardizing agencies shows such a wide diversity of the standards being set up that one wonders whether in time any article or product will escape their purview. One may enter his butcher's shop and, armed with a set of standard colors prepared by the Bureau of Animal Industry, determine whether the beef on sale is of the proper hue. Or if one wishes to arrange specimens of insects in standard vases, the National Museum has prepared the standard. If one wishes to breathe a standardized air, consult report No. 218 of the National Advisory Committee for Aeronautics entitled Standard Atmosphere—Tables and Data. Between these extremes these Federal agencies are standardizing innumerable products which vitally affect the everyday life of the average citizen.

Whether or not the said average citizen is aware of the existence of the standard or is conscious of any profit accruing to himself therefrom, does not in any way lessen the actual benefit received. Standardization reduces manufacturing costs, reduces middlemen's costs, and reduces selling costs. This means that makers, distributors, and users are all interested financially in the subject. That this is so is evidenced by the fact that 205 organizations representing makers, dealers, and users are shown in this book to be actively identified with the movement. The character and extent of these activities are outlined in a concise and readable manner and in sufficient detail to give a general view of the standardization work of each of these organizations. With all of these agencies at work, some of them organized and existing solely or largely for work on standardization, for example, the American Society for Testing Materials, it is no wonder that the progress is rapid.

CLOYD M. CHAPMAN.[3]

Books Received in the Library

AIRCRAFT YEAR BOOK, 1927. Aeronautical Chamber of Commerce of America, New York, 1927. Cloth, 6 × 9 in., 396 pp., illus., diagrams, maps, $5.25.

As in previous issues, this year book supplies a review of developments in civil and military aviation during 1926, together with much statistical material and miscellaneous information of value to those at work in this field. The reports cover all phases of aviation, commercial, technical, and historical.

THE BRIDGE TO FRANCE. By Edward N. Hurley. J. B. Lippincott Co., Philadelphia, 1927. Cloth, 6 × 9 in., 338 pp., illus., portraits, $5.

Mr. Hurley became Chairman of the United States Shipping Board and, ex-officio, President of the Emergency Fleet Corporation soon after the United States entered the war, and continued in office until the middle of 1919. Faced with the tremendous task of acquiring and operating sufficient cargo ships to maintain an army in France, the Shipping Board successfully surmounted the enormous difficulties before it.

Of these difficulties and of the methods taken to overcome them he gives a most interesting account. Here are the stories of the fabricated steel ship, of Hog Island, of the labor troubles, of the convoy service, and of the multitude of other problems that arose. In addition the book gives much of interest upon the naval, military, political, and economic strategy of the Allies.

Although the book contains much detail, Mr. Hurley has skilfully avoided a dry recital of statistics and technical details and has produced an eminently readable, interesting book. Interesting descriptions are given of many prominent personages with whom he came in contact, and there is much comment upon political events here and abroad. The book is a decided addition to the histories of the war.

DARSTELLENDE GEOMETRIE FÜR MASCHINENINGENIEURE. By Marcel Grossman. Julius Springer, Berlin, 1927. Paper, 7 × 10 in., 236 pp., diagrams, 15 r.m.

A college textbook which treats descriptive geometry from the point of view of the designer of machines rather than from that of the mathematician.

EXTERIOR BALLISTICS, 1926. By Ernest E. Herrmann. U. S. Naval Institute, Annapolis, Md., 1926. Cloth, 6 × 10 in., 322 pp., diagrams, tables, $6.

This book, which supersedes The Groundwork of Practical Gunnery as the textbook at Annapolis, takes account of the longer ranges of gunnery, the greater angles of departure, and the anti-aircraft gunnery which have become important within the past twelve years, and it therefore is broader in scope than its predecessor. In addition to the inclusion of the new methods required by these changes in practice, the entire text has been thoroughly revised.

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